Collated

IN TWO SECTIONS—SECTION ONE

X

Transactions

of the

A.S.M.E.

JANUARY, 1942

VOL. 64, NO. 1

Transactions

of The American Society of Mechanical Engineers

Published on the tenth of every month, except March, June, September, and December

---

Published monthly by The American Society of Mechanical Engineers. Publication office at 20th and Northampton Streets, Easton, Pa. The editorial department is located at the headquarters of the Society, 29 West Thirty-Ninth Street, New York, N. Y. Cable address, "Dynamic," New York. Price $1.50 a copy, $12.00 a year; to members and affiliates, $1.00 a copy, $7.50 a year. Changes of address must be received at Society headquarters two weeks before they are to be effective on the mailing list. Please send old as well as new address. . . . By-Law: The Society shall not be responsible for statements or opinions advanced in papers or . . . printed in its publications (B13, Par. 4). . . . Entered as second-class matter March 2, 1928, at the Post Office at Easton, Pa., under the Act of August 24, 1912. . . .

Modern Steam Passenger Locomotives—Research and Design

By P. W. KIEFER,[1] NEW YORK, N. Y.

This paper traces the progress of development of passenger locomotives on the New York Central System from the heavy Pacific type of 1925, to the latest combination passenger and freight design of 1940, showing the continued demand for increased power and the methods used to satisfy the requirements. A brief résumé of actual results obtained is presented and in conclusion, an expression of views is offered on possibilities for further improvement in the near future without departure from the conventional reciprocating design.

IN RECENT years, much valuable information and data on the modern steam locomotive, its elements and adjuncts, have been published or otherwise made available by many eminent engineers interested in railroad motive power. As a result of these painstaking efforts, the subject as a whole and the factors into which it subdivides have been particularly well covered. Nevertheless, when dealing with the subject of motive power from the standpoint of meeting the constantly changing needs of the modern railroad, even over a relatively short period of time, it soon becomes evident that the problems to be faced and solved are both complex and numerous and usually urgent.

With these circumstances indicated, it will be the author's main purpose to review the practical aspects of the problems which have been involved in supplying high-capacity steam motive power for fast and heavy service on the New York Central System during the last 15 years; to recite the several design and performance objectives, as determined in advance for this motive power; to submit information indicating the extent to which these objectives have been attained, as checked through performance tests under regular passenger-train operating conditions; and, finally, to express some thoughts on trends for the near future in design improvement.

While many of the problems confronted are similar to those encountered on other large railroads and the solutions in a number of respects are alike, it is desired to discuss specific matters which can be directly handled and to avoid the use of generalities so far as practicable. For these reasons, the content of this paper has been confined to the motive power of the System with which the author is connected.

It is not the intention to detail the features of the motive-power units here included but instead to limit the descriptive matter to major characteristics of design, supplemented in some instances by reference to special equipment items and to avoid, as far as practicable, repetition of what is already well known or is readily procurable from various sources.

Historical Background

In 1904, when the "Consolidation," "Ten-Wheeler," "Atlantic," and "Prairie" types were still the conventional freight and passenger locomotives in common use for heavy duty on the System, the first of a series of "Pacifics," class K-80, was introduced on the Michigan Central Railroad. During the following year, passenger locomotives of the same type were placed in service on the Boston & Albany and on the C.C.C.& St.L. Railroads. In 1907, the New York Central and the Lake Shore and Michigan Southern, now New York Central Lines West, received modifications of this type in the form of somewhat larger locomotives, designated as class K-2.

These engines successfully handled the work assigned to them and succeeding lots of the same type were installed until 1911, when a somewhat heavier and more powerful Pacific type was produced. This design is known as the class K-3, a considerable number of which are still in active service.

Continuing Demand for Increased Power

Shortly after the introduction of the last of the K-3 class, it became evident that a further substantial increase in power was required and an attempt was made to meet this demand by a yet larger Pacific, having 25-in. × 28-in. cylinders instead of 23 1/2-in. × 26-in., and with firebox and boiler capacity increased proportionately. With 79-in-diam driving wheels and a working boiler pressure of 200 psi, these locomotives developed a rated main-engine tractive effort of 37,650 lb which, by the use of a booster, was increased to 47,350 lb. This design was designated as class K-5, and, in view of its increased size, hand firing was no longer practicable for capacity operation, so mechanical stokers were installed.

The progressive development of the Pacific type from the original class K-80 on the Michigan Central, built in 1904, to the latest class K-5, built in 1926, is shown in Figs. 1 to 5, inclusive, the principal characteristics of each typical design being indicated. Fig. 6 shows the drawbar pull and drawbar horsepower versus speed for each of these distinct designs, the curves being typical of actual performance on the road under regular operation with locomotives in good condition.

The K-5 class, built in 1925 and 1926, was supplied with a tender carrying 15,000 gal of water on two 6-wheel trucks, which marked the introduction of the large-capacity tender for System passenger operation, the progressive elimination of service stops, and the extension of locomotive runs.

A survey undertaken in 1926 of the facts and conditions as related to the necessity for a further increase in the power of main-line passenger locomotives, together with consideration of probable future needs, led quickly to the definite conclusion that a unit of an entirely new design and type must be developed.

A New Type Locomotive

The basic problem presented was to create a design of locomotive having the following characteristics, as compared with the Pacific types heretofore used:

1 Somewhat greater starting tractive effort with increased horsepower capacity and maximum output at much higher speed.

2 Boiler of ample sustained capacity to satisfy the cylinder requirements for maximum power development, under severe weather and other conditions.

3 Weight distribution, wheel loads, and counterbalance to be such that impact forces and rail stresses could be confined to

---

[1] Chief Engineer, Motive Power and Rolling Stock, New York Central System. Mem. A.S.M.E.

Contributed by the Railroad Division and presented at the Semi-Annual Meeting, June 16–19, 1941, Kansas City, Mo., of The American Society of Mechanical Engineers.

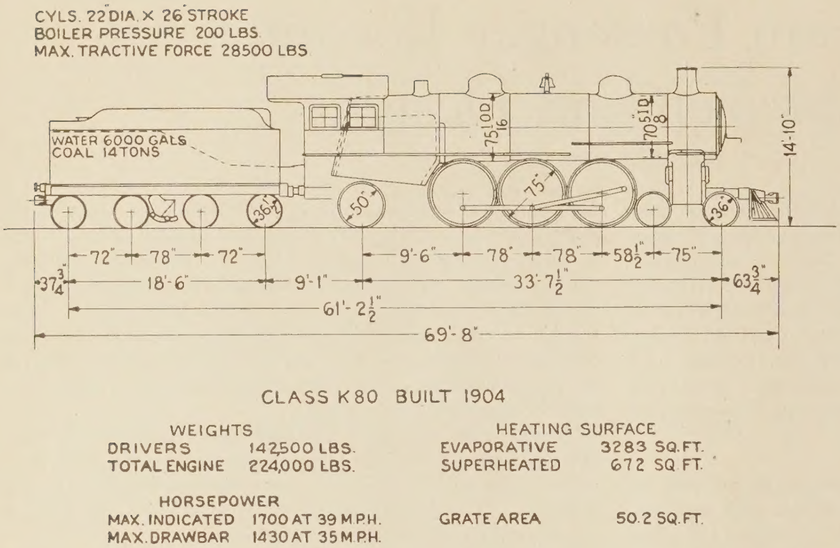


Fig. 1 Michigan Central Class K-80 Pacific Type Locomotive, Built 1904

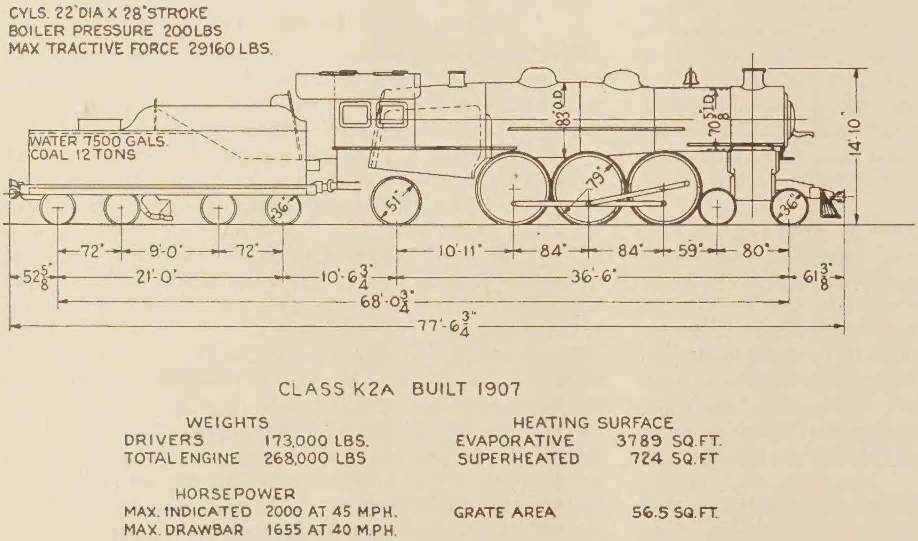


Fig. 2 Class K-2a Locomotive, Built 1907

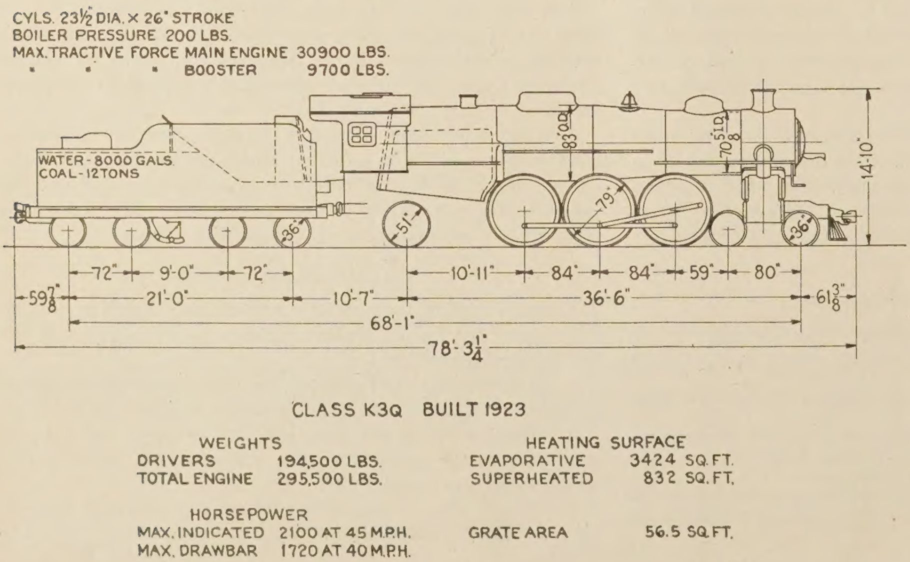
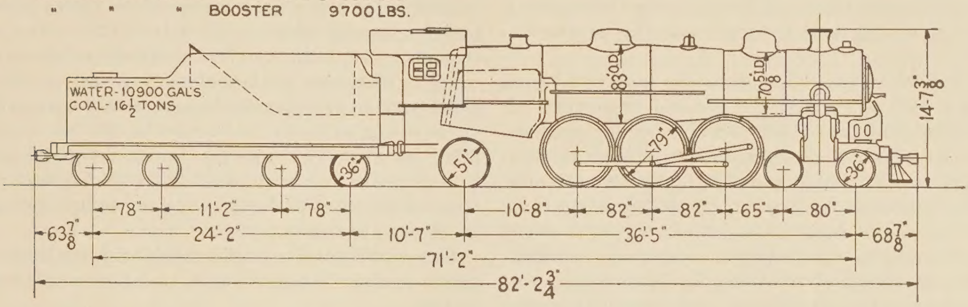


Fig. 3 Class K-3q Locomotive, Built 1923

CYLS. 24" DIA. X 26" STROKE
BOILER PRESSURE 200 LBS
MAX. TRACTIVE FORCE MAIN ENGINE 32200 LBS.
" " " BOOSTER 9700 LBS.



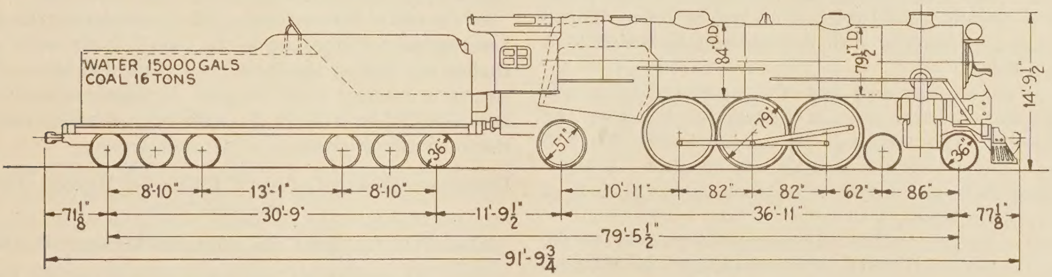
CLASS K3R BUILT 1925

WEIGHTS		HEATING SURFACE	
DRIVERS	169,000 LBS.	EVAPORATIVE	3421 SQ.FT.
TOTAL ENGINE	278,000 LBS.	SUPERHEATED	839 SQ.FT.

HORSEPOWER			
MAX. INDICATED	2140 AT 45 M.P.H.	GRATE AREA	56.7 SQ.FT.
MAX. DRAWBAR	1750 AT 40 M.P.H.		

Fig. 4 Class K-3r Locomotive, Built 1925

CYLS. 25" DIA. X 28" STROKE
BOILER PRESSURE 200 LBS.
MAX. TRACTIVE FORCE MAIN ENGINE 37650 LBS.
" " " BOOSTER 9700 LBS.



CLASS K5B BUILT 1926

WEIGHTS		HEATING SURFACE	
DRIVERS	185,000 LBS.	EVAPORATIVE	3952 SQ.FT.
TOTAL ENGINE	302,000 LBS.	SUPERHEATED	1150 SQ.FT.

HORSEPOWER			
MAX. INDICATED	3200 AT 54 M.P.H.	GRATE AREA	67.8 SQ.FT.
MAX DRAWBAR	2530 AT 45 M.P.H.		

Fig. 5 Class K-5b Locomotive, Built 1926

lower limits than heretofore observed, thus contributing to higher standards of track maintenance and obtaining better train riding characteristics.

4 Increased thermal efficiency.

5 Clearances to permit operation without restriction on the various parts of the System.

6 Symmetrical appearance with smooth lines, free from the effects of miscellaneous appliances, piping, and other details.

7 A high degree of reliability for uninterrupted service under conditions of dense traffic, especially on the eastern section of the System, requiring relatively simple but adequate machinery, combined with the use of well proved auxiliary equipment, such as feedwater heaters and mechanical stokers.

After the preparation of several preliminary designs, in which the American Locomotive Company, the Superheater Company, and others cooperated to the fullest extent, the conclusion was reached that the objectives could be most efficiently attained by using a 4-6-4 wheel arrangement which would satisfy the requirements for capacity and weight and avoid the addition of a fourth

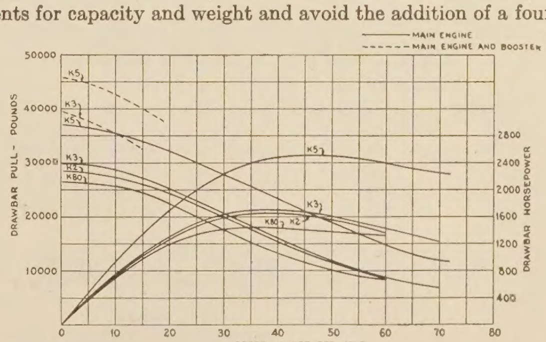


Fig. 6 Curves Showing Drawbar Pull and Horsepower Versus Speed for Class K Locomotives

pair of driving wheels, with a resultant increase in size, weight, and first cost, as well as higher maintenance costs.

This arrangement represented the first 6-driver locomotive, built with 4-wheel leading and trailing trucks for service in America.

To meet the demand for exceptional steaming capacity at sustained high speed with heavy load, the size and proportions of the boiler were given first consideration, ample heating surfaces being essential, with extra-large superheater and a grate area sufficient to insure an economical rate of firing under maximum conditions of steam generation. To carry the added weight thus imposed on the rear of the locomotive, without excessive loads on trailing or coupled axles, the 4-wheel trailing truck was used, thus securing the advantage of providing for large firebox capacity with comparatively light individual axle loads and consequent low rail stresses. The boiler as finally designed had the following general dimensions and proportions:

Firebox 130 in. long $\times$ $90^{1}/_{4}$ in. wide
Volume of firebox, 428 cu ft
Diameter of boiler at smokebox, $82^{7}/_{16}$ in.
Diameter of boiler at back tube sheet, $87^{5}/_{8}$ in.
Tubes, 201–$3^{1}/_{2}$ in. and 37–$2^{1}/_{4}$ in., 20 ft 6 in. long
Heating surface; tubes, 4203 sq ft
Heating surface; firebox, 281 sq ft
Heating surface; total, 4484 sq ft
Superheater, type E; heating surface 1951 sq ft
Gas area through flues and tubes, 9.67 sq ft
Grate area, 81.5 sq ft

In developing the boiler design, the provision of a combustion chamber was carefully considered but, because of serious difficulties then being experienced with riveted-seam construction, it was finally omitted.

To reduce the pressure drop and other losses and to provide for more efficient use of the steam in the cylinders, the steam and exhaust passages were enlarged, as compared with the K-5 Pacific type, and a front-end throttle was installed. A large-volume steam chest with valves 14 in. diam, similar to those of the K-5 type, was retained.

Other special features included air compressors mounted on the front deck for improved weight distribution and, for the first time, a specially designed cast-steel pilot and drop coupler, providing a surface free from the projection of coupler and pocket for clearing effectively possible obstructions on the right of way. The centrifugal-type boiler feed pump was first introduced on this design with the heater located in a recessed portion of the smokebox top and a large portion of the piping placed under the jacket. Careful attention was given to the arrangement of controls and gages in the cab for convenient access and clear vision.

Fig. 7 illustrates the last of the J-1 class as received in 1931 and also shows the principal engine dimensions and proportions of the design as finally determined for the first sample J-1A, No. 5200,[2] built in 1927, except that the weight on drivers was 182,000 lb and total engine weight was 343,000 lb. A tender with 4-wheel trucks was used with this first engine having a capacity of 10,000 gal of water and 18 tons of coal.

From 1927 to 1931, a total of 205 of these locomotives, designated as the "Hudson" type, were received and placed in service.

Subsequently, all of the J-1 class were dynamically counterbalanced to provide smoother operation and to permit the use of shorter running cutoff, as well as to improve the track effects. Roller bearings were installed on all engine and tender trucks and to the drivers of eight locomotives. All engines received speed recorders, later augmented by cutoff-selection equipment.

Cast-steel beds with integral cylinders were applied, the engines already being equipped with one-piece cast-steel tender frames, engine-truck, trailer-truck, and tender-truck frames. The substitution of integral construction for the multiple-bolted parts of earlier locomotives eliminated a large number of bolts and contributed to increased availability and continuity of operation with substantial reduction in maintenance costs.

At the end of 1940, a total of 437 locomotives of this 4-6-4 type had been placed in service in the United States and Canada, including the 275 on the New York Central System. The total weight in working order for each of these locomotives ranged from 310,000 to 415,000 lb, with corresponding variations in maximum tractive force.

Performance and Capacity Tests, J-1 Hudson Versus K-5 Pacific Type

Class J-1A No. 5200 was subjected to complete performance

[2] "First Hudson Type Locomotive," *Railway Age*, vol. 82, 1927, pp. 523–526.
"Hudson Type Locomotive on N.Y.C.," *Railway Mechanical Engineer*, vol. 101, 1927, pp. 139–141.

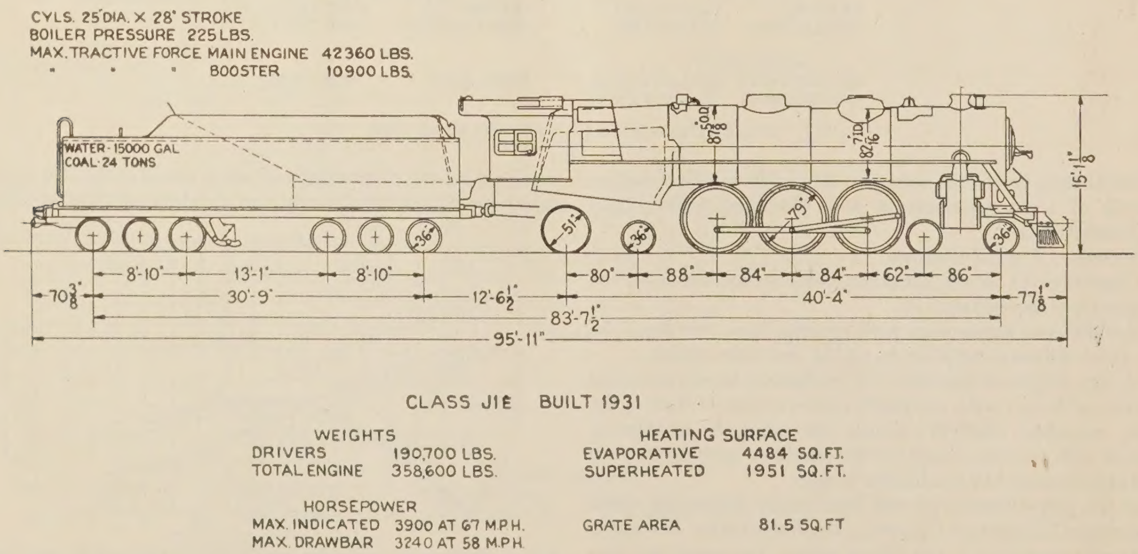


Fig. 7 Class J-1E Locomotive, Built 1931

and capacity tests shortly after delivery in 1927. Because of the total engine weight being held to 343,000 lb and the smaller and lighter tender used, this locomotive delivered a maximum drawbar horsepower of 3300 at 58 mph. However, subsequent improvements already referred to increased the weight of the Hudson-type locomotives and, consequently, the principal test results here given are for the last-built and heavier class J-1E tested in 1937.

The complete performance and capacity tests of classes J-1E (No. 5339) and K-5B (No. 8363) were conducted under spring and summer weather conditions over the Mohawk Division of the New York Central between Albany and Syracuse, a distance of 140 miles. This division is generally representative in profile and operating characteristics of the main line between New York and Chicago with the exception of the severe though comparatively short grade westbound between Albany and West Albany, a distance of about 3 miles, where the maximum grade is 1.63 per cent on a curvature of $3^1/_2$ deg. With a total rise westbound of 384 ft in the 140-mile division over a rolling profile, the average grade is 0.05 per cent with a maximum of about 0.5 per cent for approximately 1.5 miles westbound and about 0.75 per cent for slightly over 2 miles eastbound.

All tests were made under regular-road-service conditions of operation, the trains consisting of empty standard steel passenger coaches varying in number from ten to twenty which, with a dynamometer car, provided train weights of 780 to 1465 tons. These trains were selected as representative of normal daily operation expected of the locomotive. Average test results demonstrated that the class J-1 Hudson type surpassed all previous New York Central locomotives in maximum horsepower, coal and water consumption per horsepower, weight per horsepower, and over-all efficiency.

A comparison of the principal results obtained for a single division run with representative trains is given in Table 1. It should be especially noted that, except for the maximum-power characteristics which may be duplicated at will with full boiler pressure and locomotive in good condition, the results shown are on the basis of over-all averages for the complete division runs, and indicate regular daily service performance rather than maximum values for short periods or under controlled conditions for the separate items.

TABLE 1 COMPARISON OF TEST RUN FOR K-5 AND J-1 LOCOMOTIVES

	Maximum power K-5	J-1	Improvement, J-1, per cent
Tractive effort with booster, lb	48750	55100	13.0
Main-engine tractive effort, lb	40000	45400	13.5
Main-engine drawbar pull, lb	37000	41300	11.6
Cylinder horsepower	3200	3900	22.0
	at 54 mph	at 67 mph	24.1
Drawbar horsepower	2530	3240	28.1
	at 45 mph	at 58 mph	28.9
	Average-performance data		
Number of cars and weight in tons	15–1053	18–1244	..
Average working speed, mph	51.2	55	..
Average firing rate, lb dry coal per hr	5867	6940	..
Water delivered to boiler, lb per hr	40636	57200	..
Evaporation per lb of dry coal, lb	6.94	8.24	18.7
Combined efficiency—boiler, feed-water heater and superheater, per cent	67.8	74.6	10.0
Steam per indicated horsepower-hour, lb			
Cylinders only	15.42	15.44	..
Including auxiliaries	17.00	17.28	..
Dry coal per indicated horsepower-hour, lb			
Cylinders only	2.22	1.94	12.6
Including auxiliaries	2.46	2.10	14.6
Coal as fired per car mile, lb	7.22	7.03	2.6
Weight per indicated horsepower, lb	94	90	4.3
Based on working-order weight, lb	302000	352000	..

Comparative curves representing the drawbar pull and drawbar horsepower versus speed are shown in Fig. 8, which also includes curves for locomotives of more recent design as discussed later. With a starting effort approximately 12 per cent greater than the K-5, increasing to 37 per cent more at a speed of 70 mph, and with an increase of 28 per cent in maximum drawbar horsepower at a speed 29 per cent higher, the weight per horsepower of the J-1 Hudson type has been decreased.

The Improved Hudson Type, Class J-3

As early as 1931, when the last of the J-1 class was built, consideration was already being given to the future development of this type in anticipation of greater power demand necessitated by the constantly increasing weight of trains and shortening of schedules. In order to reduce weight and also to gain some experience in the use of alloy steel of high tensile strength, with a view toward increasing the steam pressure, three of these locomotives were equipped with nickel-steel boilers. Two of the three also had roller bearings on all wheels except on the trailing truck, and the entire lot had roller bearings on the engine truck and tender wheels.

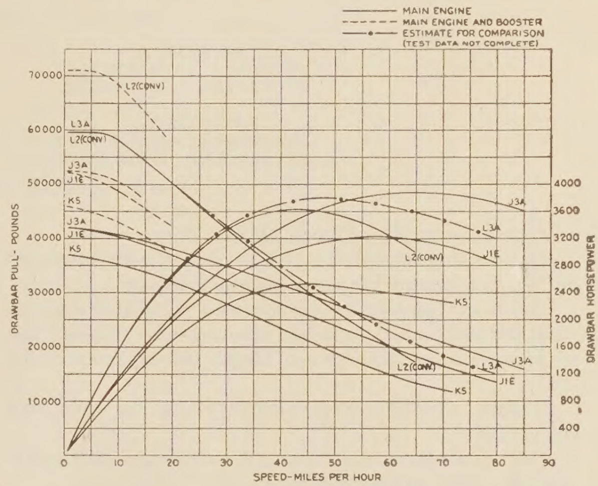


FIG. 8 CURVES SHOWING DRAWBAR PULL AND HORSEPOWER VERSUS SPEED FOR VARIOUS TYPES OF LOCOMOTIVES

Subsequently, one of the three, No. 5344, received lightweight roller-bearing rotating and reciprocating parts and the counterbalance was reduced proportionately, providing lower rail stresses and improved riding qualities. At this time, the boiler pressure was raised from 225 to 250 psi and the cylinders were bushed to preserve the same starting tractive effort and adhesion factor as on others of the same class.

As previously explained, successive lots of the J-1 class had received various improvements when built and subsequently with gradual increase in the weight of engine and tender. The weights of the original class J-1A and the latest class J-1E compare as follows:

	Original J-1A	Latest J-1E
Engine truck, lb	63500	65700
Drivers, lb	182000	190700
Four-wheel trailing truck, lb	97500	102200
Total engine, working order, lb	343000	358600
Tender, fully loaded, lb	212200	305600

On the basis of the J-1 test results, the experience accumulated with the altered locomotives of this class, and other considerations, the general objectives for the new design were set as follows:

1 Maximum cylinder horsepower approximately 20 per cent greater at a much higher speed.

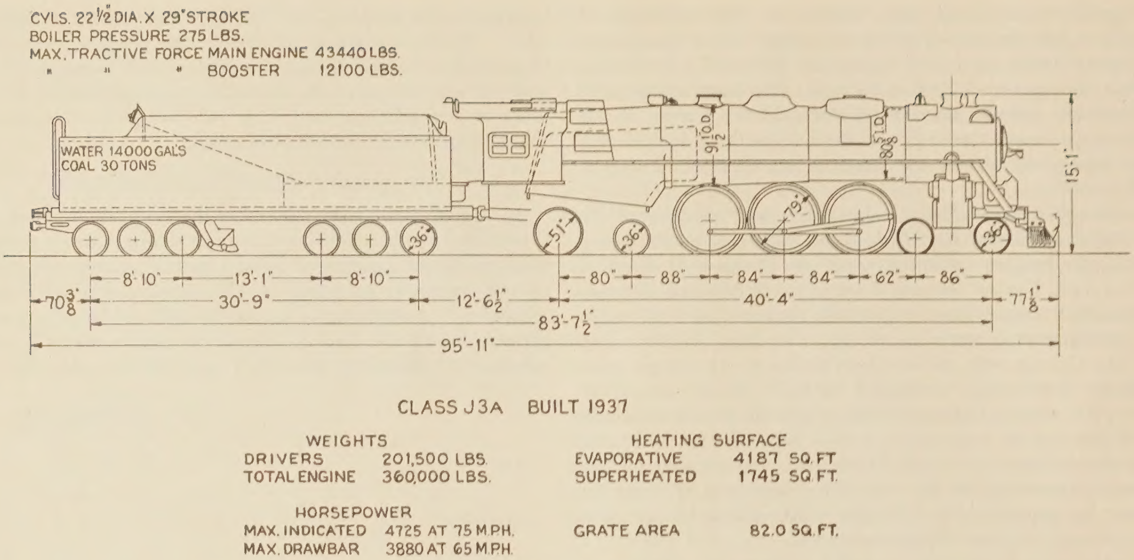


Fig. 9 Class J-3A Locomotive, Built 1937

2 Boiler pressure 275 psi versus 225 psi.

3 Equal main-engine starting tractive effort, with some additional help from booster because of increased pressure.

4 Boiler and superheater proportioned for higher capacity demand and to insure ample supply of steam under all conditions.

5 Approximately same over-all length and clearance limitations.

6 Highest capacity tender possible within the then total length limitation.

7 Least possible increase in weight, and weight distribution no less favorable from track standpoint.

Careful study of the situation indicated that, with the utmost attention to all details of design, these objectives could be attained and still adhere to the 4-6-4 wheel arrangement rather than using another pair of driving wheels, thus effecting substantial savings in size, weight, first cost, and operating expense.

In the development of the new design, the cooperation given by The American Locomotive Company, the Superheater Company, the Timken Roller Bearing Company, and others was of the utmost value.

Fifty of these locomotives were built in the fall of 1937 and the spring of 1938, ten of which were streamlined and five of these had roller bearings on main and side rods. The principal dimensions and proportions[3] are shown in Fig. 9.

As indicative of the design characteristics of the boiler, without elaboration of the details, the following information is given:

Material, nickel steel
Working pressure, 275 psi
Firebox, $130^{13}/_{16}$ in. long $\times$ $90^{1}/_{4}$ in. wide
Volume of firebox, 519 cu ft
Diameter of boiler at smokebox, $80^{5}/_{8}$ in.
Diameter of boiler at back tube sheet, $91^{1}/_{2}$ in.
Combustion chamber, 43 in. long
Tubes, 183–$3^{1}/_{2}$ in. and 59–$2^{1}/_{4}$ in., 19 ft long
Heating surface; tubes, 3827 sq ft
Heating surface; firebox, 360 sq ft
Heating surface; total, 4187 sq ft
Superheater, type E; heating surface 1745 sq ft
Gas area through flues and tubes, 8.91 sq ft
Grate area, 82 sq ft

[3] "New N. Y. C. Locomotives Show High Power Concentration," *Railway Age*, vol. 104, 1938, pp. 597–605.

"The New York Central Receives Fifty Powerful 4-6-4 Locomotives," *Railway Mechanical Engineer*, vol. 112, 1938, pp. 165–173.

With a boiler pressure of 275 psi, cylinder sizes of $22^{1}/_{2}$ in. diam and 29 in. stroke were fixed to produce a main-engine starting tractive effort of 43,440 lb, slightly more than the 42,360 lb of the J-1 class. The booster provided an additional 12,100 lb starting effort. A large-volume steam chest with 14-in-diam valves, similar to the J-1 class, was retained but the steam passages from dome to exhaust were enlarged in proportion to the cylinder area to provide free passage of the steam and reduce losses in transmission.

Special design and equipment features were as follows:

Roller bearings applied to all wheels.
Reciprocating parts of special lightweight design.
Revolving parts reduced in weight.
Dynamic counterbalancing.
Reverse-gear cylinder located on center line of engine to assist in reducing irregularities or inequalities in valve travel due to deflection or other causes.
Speed recorder and cutoff-selection equipment.
Rubber twin-cushion double-acting draft gear at rear of tender to eliminate free slack in both directions of gear movement substituting controlled resiliency to obtain smooth and efficient operation of trains. The ten streamlined engines received tight-lock couplers.

The requirements for increased cylinder power and consequent greater boiler capacity and higher working steam pressure, together with the roller-bearing equipment, improved brakes, additional sand-box capacity, and certain minor items, indicated a weight increase of about 14,750 lb over that of the latest class J-1E, but, as previously stated, one of the major objectives was to hold the weight as closely as possible to that of the J-1 class and to accomplish this the following features were incorporated in the design:

Nickel-steel boiler-shell sheets.
Cast-steel unit-bar grates.
High-tensile-steel drop coupler.
Cor-Ten steel main air reservoirs.
Aluminum cab, running boards, casings, and gage board.
Magnesia-block lagging of light weight.
Tubes and flues to close tolerance.
Booster exhaust piped to tender instead of to stack.

Integral cast-steel frames and cylinders, cradle, engine-truck and trailing-truck frames of lightened design.

Lightweight new-design valve gear.

Lightweight reciprocating parts and alloy-steel rods also contributed to the saving in weight.

The resulting weight reduction amounted to 13,350 lb, making the net addition only 1400 lb with a total weight of engine in working order of 360,000 lb, of which 201,500 lb were placed on the drivers.

With this total weight and the distribution obtained, together with the use of reduced-weight rotating and reciprocating parts and dynamic counterbalancing, the calculated stresses on the track structure were satisfactory and well within permissible limits.

Performance and Capacity Tests, J-3 Versus J-1

The tests of the J-3 were conducted with engine No. 5408 during the last 3 months of 1937, over the Mohawk Division under regular service conditions of operation, the trains consisting of 22, 17, and 10 cars, which with the dynamometer car, furnished weights back of the tender of 1609, 1244, and 766 tons, or heavy, medium, and lightweight trains.

TABLE 2 PERFORMANCE OF J-3 CLASS LOCOMOTIVE COMPARED WITH J-1 CLASS

	Maximum power J-1	Maximum power J-3	Improvement, J-3, per cent
Tractive effort with booster, lb	55100	55000	..
Main-engine tractive effort, lb	45400	45000	..
Main-engine drawbar pull, lb	41300	41500	..
Cylinder horsepower	3900	4725	21.1
	at 67 mph	at 75 mph	11.9
Cylinder horsepower per pair of driving wheels	1300	1575	21.1
Drawbar horsepower	3240	3880	19.75
	at 58 mph	at 65 mph	12.1
	Average performance, division run of 140 miles		
Number of cars and weight in tons	18–1244	18–1253	
Working speed, mph	55	59	
Firing rate, dry coal per hour, lb	6940	6419	
Water delivered to boiler per hour, lb	57200	54900	
Evaporation per pound of dry coal, lb	8.24	8.32	1.0
Combined efficiency; boiler, feedwater heater, and superheater, per cent	74.6	76.3	2.3
Steam per indicated horsepower-hour, lb			
Cylinders only	15.44	14.76	4.4
Including auxiliaries	17.28	16.89	2.3
Dry coal per indicated horsepower-hour, lb			
Cylinders only	1.94	1.84	5.15
Including auxiliaries	2.10	2.03	3.3
Coal fired per car mile, lb	7.03	6.21	11.7
Weight per indicated horsepower, lb	90	76	15.5
Based on weight of engine in working order, as tested, lb	352000	360000	

The principal results of representative performance are given in Table 2, the figures for the class J-1 being repeated for ready comparison.

The drawbar pull and drawbar horsepower throughout the speed range are included in Fig. 8, with other types for comparison. Fig. 10 also shows the cylinder tractive effort and horsepower for the J-3A class only.

While the same main-engine starting tractive effort has been obtained in the new design, as desired, the drawbar pull at 70 mph has increased nearly 25 per cent, and the maximum drawbar horsepower is 20 per cent greater at a speed 12 per cent higher than the J-1. Coal and steam consumption per horsepower-hour have been decreased with a reduction of 15 per cent in weight per horsepower. An average thermal efficiency of 6.06 per cent at the drawbar was obtained for a complete division run, corresponding to 9.6 per cent at the cylinder.

Thermal Efficiency at Tender Drawbar Referred to Fuel

Reference to this value for the conventional-design steam locomotive usually affords an opportunity for considerable argument, although it is a fact that, during recent years, gradual improvement in this respect has been achieved.

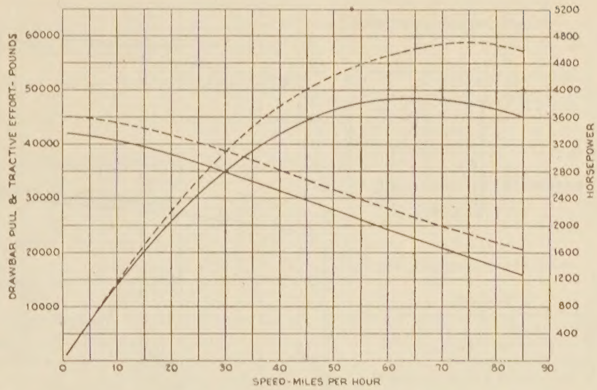


FIG. 10 TRACTIVE EFFORT AND HORSEPOWER VERSUS SPEED; AND DRAWBAR PULL AND HORSEPOWER VERSUS SPEED FOR J-3A LOCOMOTIVES

Without questioning the fact that it is highly desirable to improve this performance characteristic, it is prudent to review some of the reasons for the relatively low thermal-efficiency value, and to consider the practical advantages inherent in this form of motive-power plant.

The conventional locomotive is a noncondensing self-contained and self-propelled unit, confined within close and definite limits of weight, height, width, and in most cases length, because of operating clearance and load limitations. The necessarily high horsepower requirement naturally is accompanied by a high combustion rate and Btu heat release per cubic foot of firebox volume per hour. Furthermore, this complete power plant, including all auxiliary equipment and its own fuel and water supply, is handled successively by different crews of only two men each at high speeds in dense traffic under widely and rapidly fluctuating load requirements.

Steam-locomotive efficiency at the tender drawbar is affected by the non-power-producing wheels and by the weight carried thereby. The modern tender when fully loaded may represent the equivalent of $1^1/_2$ loaded 70-ton coal cars or more. However, the hauling of this nonadhesive weight is amply justified through sustained power output and the attendant advantages obtained.

Simplification of design, particularly with respect to cylinders and valve gear, penalizes the thermal efficiency, but repayment is secured and augmented in terms of high serviceability and reasonable freedom from excessive maintenance troubles. Moderate first cost for an active motive-power unit is essential unless the net return on additional investment can be clearly established. The measure of the value of a locomotive is its use, idle motive power representing a total loss of investment and a constant expense. It is currently demonstrated that economically, maximum over-all-performance efficiency is secured through the use of a unit capable of providing uninterrupted service and consistently high mileage throughout its life, with the best available design of boiler, cylinder, and related parts to fulfill these conditions.

Slipping Tests

As previously noted, the J-3 class was equipped with roller bearings throughout, including the driving wheels. The possibility was recognized that forced vibrations of the unsprung mass of the closely fitted roller-bearing driving-wheel assemblies caused by the overbalance and the elastic foundation of the track structure might be sufficient during high-speed slipping to cause

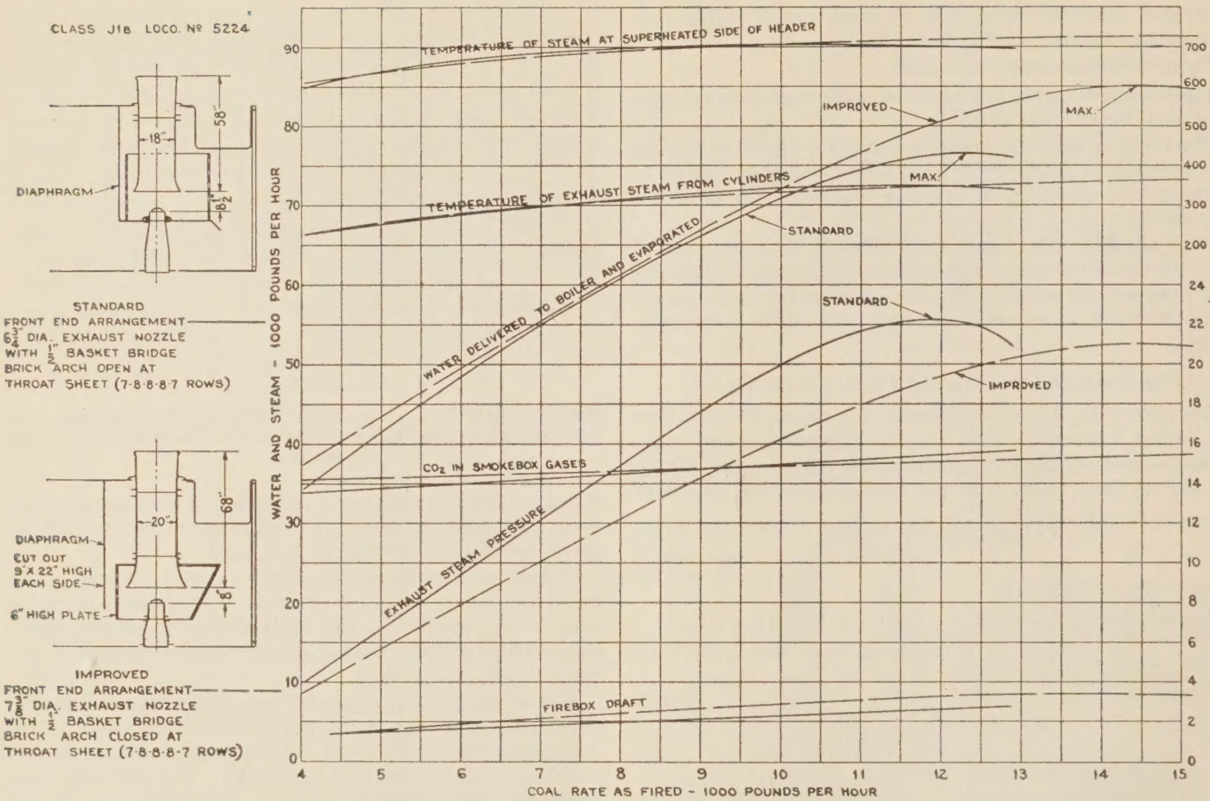


Fig. 11 Performance Curves for Standard and Improved Front-End Arrangements, as Applied on J-1 Locomotives

the wheels to lift from the rail. Freedom from lost motion and smooth-running machinery are highly desirable characteristics of this type of driving-wheel-and-axle assembly, and in order definitely to determine (1) the rotational speed at which the driving wheels actually lift from the rail, and (2) whether this speed would approach or possibly fall below the maximum recorded slipping speed of 120 mph on this class of engine, a program of slipping tests was formulated and conducted in April, 1938, on a short stretch of main-line track with a train of eight cars for trailing load. The rail was 127-lb section on rock ballast as later described under "Track Tests, L-2D Converted."

To promote slipping, the heads of both rails were greased before each run for a distance of 230 ft. Scratch gages to measure rail deflection and movement of driving boxes with respect to frame were located at marked positions on the rail and on the locomotive. The train speed and the maximum revolving speed of the driving wheels during slips were obtained by a positively driven electrical speed indicator of the Weston generator type, and high-speed motion-picture cameras were used to record any lifting of wheels from the rail.

Four test runs were made at train speeds varying from 61 to 82 mph and with maximum slipping speeds of 123, 130, 135, and 164 mph while working steam. In the three tests at the lower speeds, there were no indications that the wheels had lifted. In the final test at 164 mph, the main drivers definitely left the rail and later examination disclosed a number of very slight markings on the rails which were without significance and had no disturbing effect on the track structure requiring attention of maintenance forces.

No damage to the locomotive occurred in any of these tests and the two questions postulated were definitely answered because the rotational speed of 164 mph necessary to lift the wheels from the rail exceeded by 44 mph the highest known slipping speed of these engines.

Subsequent Improvements in Boiler Capacity and Engine Efficiency

In the year 1937, a series of standing tests was undertaken to determine the extent to which improvement in capacity and efficiency of the class J-1 and J-3 boilers and cylinders could be secured by redesign of the smokebox arrangement. The primary objectives were to increase the capacity of the boiler, accompanied by a reduction in back pressure, which would be reflected in increased cylinder horsepower and efficiency. Prior to this time, standing tests had been conducted and had furnished valuable information but were not conclusive because no means had been developed for reducing the temperature of the exhaust steam corresponding to the temperature drop, when mechanical work was being performed under service conditions.

In the tests under consideration, this reduction in temperature was effected and controlled through the medium of a spray of water mist introduced into the cylinders, the water being subsequently removed, so that the condition of the exhaust steam during the standing tests was made identical with the thermal conditions experienced in road service.[4]

---

[4] A complete description of the apparatus used and results obtained was presented by W. F. Collins, Engineer of Tests, New York Central System, at the 1940 Annual Meeting of the Railway Fuel and Traveling Engineer's Association. Reports were published as follows:

"Standing Locomotive Tests of the New York Central," *Railway Age*, Jan. 18, 1941, pp. 177–181, 186.

"New York Central's Standing Locomotive Tests," *Railway Mechanical Engineer*, part 1, Feb., 1941, pp. 56–59; part 2, March, 1941, pp. 96–100.

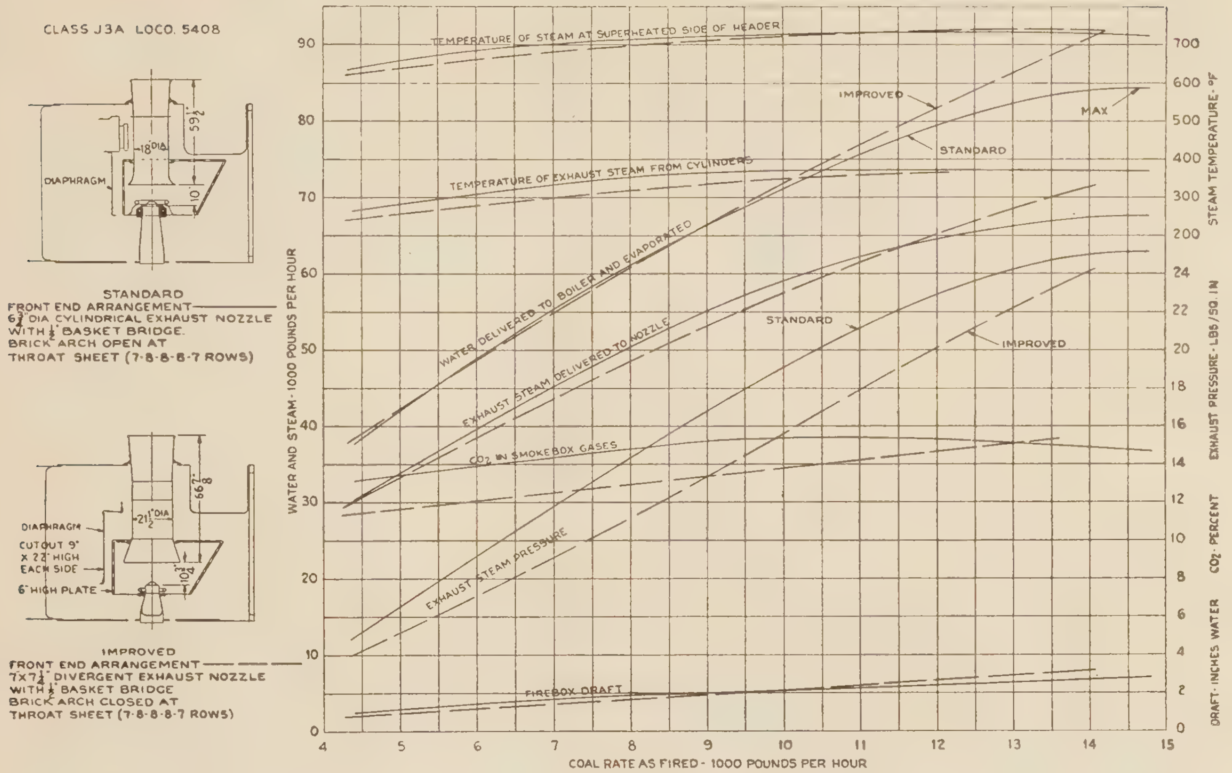


Fig. 12 Performance Curves for Standard and Improved Front-End Arrangements, as Applied on J-3 Locomotives

Several different smokebox arrangements were tested during these experiments, which were first conducted on J-1 No. 5224 at the Selkirk Engine Terminal. The design finally approved as producing the best performance is shown in Fig. 11, the previous standard arrangement also being shown, together with curves illustrating the comparative performance. Fig. 12 shows similar information for the J-3 class.

TABLE 3 SMOKEBOX TESTS FOR J-1 AND J-3 LOCOMOTIVES AT MAXIMUM EVAPORATION RATE

Item no.		J-1 Class		J-3 Class	
		Original	Improved	Original	Improved
1	Coal fired per hour, lb....	12300	14500	14700	14200
2	Coal fired per square foot of grate area per hour, lb...	151	178	179	173
3	Water evaporated per hour, lb..................	76800	85500	84500	92000
4	Increase, Item 3, per cent.	..	11.3	..	8.9
5	Evaporation per pound of coal fired, lb..........	6.24	5.89	5.76	6.48
6	Exhaust passageway pressure, psi..............	22.3	21.4	25.3	24.5
7	Decrease, Item 6, per cent.	..	6.4	..	3.2
8	Firebox draft, in. of water	2.7	3.4	2.8	3.2
9	Combined efficiency, boiler and superheater (dry coal basis), per cent....	56.8	52.6	53.0	56.0
10	Diameter of basket-bridge exhaust nozzle, in.......	6 3/4	7 3/8	6 3/4	7

TABLE 4 SMOKEBOX TESTS FOR J-1 AND J-3 LOCOMOTIVES, AVERAGE VALUES

Item no.		J-1 Class		J-3 Class	
		Original	Improved	Original	Improved
1	Coal fired per hour, lb.....	7131	7131	7175	7175
2	Coal fired per square foot of grate area per hour, lb...	87.5	87.5	87.5	87.5
3	Evaporation per pound of coal fired, lb..........	7.87	7.90	7.89	7.82
4	Exhaust pressure, psi......	12.5	10.3	12.0	9.3
5	Decrease, Item 4, per cent.	..	17.6	..	22.5
6	Firebox draft, in. of water	1.8	2.2	1.6	1.4
7	Combined efficiency, boiler and superheater (dry coal basis), per cent.....	68.5	70.0	69.4	67.8
8	Combustion efficiency (dry coal basis), per cent.....	86.0	87.8	86.0	88.1
9	Diameter of basket-bridge exhaust nozzle, in.......	6 3/4	7 3/8	6 3/4	7

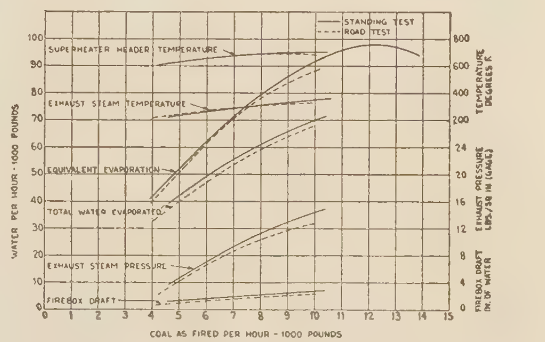


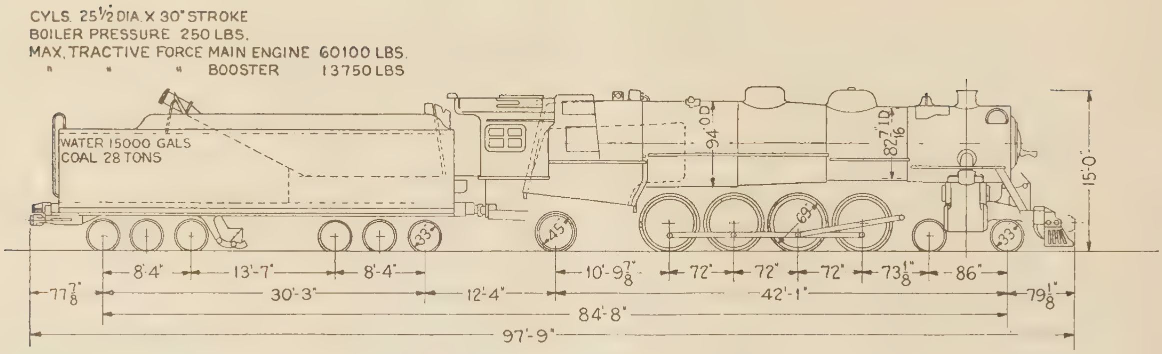
Fig. 13 Comparative Results of Standing Tests Versus Road Dynamometer Tests for Class J-1, Engine No. 5224
(Original front end, 7 3/8-in. nozzle.)

Tables 3 and 4 outline briefly the principal results for both the J-1 and the J-3 classes, the data being given at maximum evaporation rates and also for an average firing rate of about 7100 lb per hr, corresponding to 87.5 lb of coal per hr per sq ft of grate area.

While the combined efficiency of the boiler and superheater is shown as lower for the J-1 with the improved front end, this is due to the much higher coal rate made possible with this front end. The maximum evaporation has been increased about 11 per cent, with a decrease in back pressure of 6 per cent.

For the class J-3 the combined efficiency of the boiler and superheater is increased slightly at approximately the same coal rate, and at the same time the maximum evaporation is increased almost 9 per cent with some reduction in back pressure.

The fundamental means of accomplishing these results was by



CLASS L2D (CONV.) BUILT 1930-CONVERTED 1939

WEIGHTS		HEATING SURFACE	
DRIVERS	257,000 LBS.	EVAPORATIVE	4556 SQ.FT.
TOTAL ENGINE	385,100 LBS.	SUPERHEATED	1931 SQ.FT.

HORSEPOWER			
INDICATED	4200 AT 50 M.P.H.	GRATE AREA	75.3 SQ.FT.
DRAWBAR	3640 AT 43 M.P.H.		

Fig. 14 Characteristics and Principal Dimensions of L-2d Class Locomotives, Built in 1930, and Converted in 1939

providing a smokebox arrangement which, with the proper relation of exhaust nozzle and stack, resulted in moving a greater quantity of gas with the same quantity of steam.

Under average conditions of firing, representing normal operation of the locomotive, the evaporation and efficiency of the boiler have been maintained with a reduction in exhaust pressure, which is reflected in increased cylinder horsepower and over-all thermal efficiency.

Fig. 13 has been prepared to show graphically the results from a road dynamometer test and a standing test for the same class J-1 locomotive with identical arrangement of smokebox, firebox, and exhaust nozzle, illustrating the close degree to which the standing tests duplicate the thermal conditions which occur when steam is used during road operations.

As a result of the experiments, installations of the improved design of front-end arrangement have been proceeding and approximately 520 engines are now equipped.

Conversion of Two 4-8-2 Locomotives for Passenger Service

With the full complement of 275 Hudson type on the System, it was found that the passenger traffic could be handled satisfactorily under normal conditions but that, during peak periods in the holiday seasons, it was necessary to use some of the older K-3 Pacific type for the excess traffic. As these units were gradually retired and assigned to secondary or branch-line service, there remained an insufficient number available for this supplemental operation and, as the trains handled often demanded greater power than these engines possessed, they were not satisfactory for this purpose.

For these reasons the 4-8-2 Mohawk-type freight engines, class L-2, were occasionally used in emergency passenger service during heavy-traffic periods but were limited to 60 mph due to riding qualities and the difficulty of maintaining satisfactory running conditions with the friction-bearing driving boxes.

These engines were built during the years 1926 to 1930. When the acquisition of new freight power was lately under consideration, necessitating complete review of the design, with a number of important changes, the question arose as to whether the new design could be so arranged as to preserve the general character of freight locomotives and at the same time serve satisfactorily in passenger service during periods of peak traffic. Such additions to the freight motive power would also accomplish an increase in the available passenger power without actually involving additional units, designed especially for passenger service.

Consideration was given to the design and construction of a sample locomotive to be given a thorough test and trial but, because of the time involved and the complications usually present in an undertaking of this kind, it was decided that one or two of the L-2 class should be converted for high-speed passenger service, thus making possible early development of experience background for the design of the new engines, an order for which could then be placed without undue delay.

Two of the L-2 class, Nos. 2995 and 2998, were selected for this purpose and the following principal changes were made to provide satisfactory operation in passenger service at speeds of 80 mph and at the same time maintain suitability for freight operation equal to the L-2 class:

1 Boiler pressure increased from 225 to 250 psi.
2 Cylinder diameter reduced from 27 in. to $25^1/_2$ in. for starting tractive effort equal to the L-2.
3 Lightweight reciprocating parts.
4 Dynamic counterbalancing of all drivers.
5 Roller bearings on engine truck, tender truck, and drivers on No. 2998.
6 Roller bearings on engine truck and tender truck on No. 2995.
7 Coal pushers in tender.
8 Lateral-motion device on front drivers.
9 Improved radial buffers between engine and tender.
10 Cast-steel pilots and drop couplers.

The weights before and after conversion were as follows:

Weight in working order, lb	L-2 Freight	L-2 Converted
Engine truck................	59150	65400
Drivers.....................	250000	257000
Trailing truck..............	61000	62700
Total engine................	370150	385100

The two locomotives were released for service in August, 1939, and have been successfully handling main-line passenger trains since that time, except that one of them was removed from service for exhibition at the World's Fair throughout the 1940 season. As of December 31, 1940, a total of 200,000 miles had been accumulated on the two engines and no special difficulties of

operation or maintenance have been experienced during this period of service. Fig. 14 shows the dimensions and principal characteristics of the converted engines.

Track Tests, L-2D Converted

The weight of the two converted class L-2 locomotives had been increased about 15,000 lb over the standard L-2 and it was essential to determine whether these engines, with 69-in. driving wheels and the modifications referred to, could be operated at the passenger-train speed of 80 mph without imposing excessive stresses on the track structure. For this purpose, track tests were conducted in September, 1939, which included one of the J-1 class as well as the two converted engines, in order to obtain comparative information, as the J-1 class, during approximately 200,000,000 miles of operation, had never been known to produce any harmful effects on the track.

Two 170-ft test sections were used, located about $^1/_2$ mile apart on the inside westbound high-speed main track No. 1 of a 4-track system. Both sections are on an ascending grade of 0.315 per cent, one comprising tangent track and the other a curve of 1 deg 8 min, selected so that each test was run continuously over both sections without reduction in speed for the curve.

The rail was 127-lb New York Central standard section laid on sound creosote-treated ties spaced about 1 ft 8 in. center to center with canted tie plates having an outside shoulder only. The ballast was 2-in. crushed rock.

Strain gages placed in groups at 10-ft intervals were used to measure the stresses in the rails on the outside of the rail head, underneath the rail at the center line, and on top of the outer and inner flanges. Slow-motion pictures were taken at each test section to determine the position of the crankpin for each stress recorded.

The results showed that up to 87 mph, the maximum speed operated, the converted L-2 imposed no greater stress on the track than the J-1 and that the maximum stresses in both cases were well within permissible limits, proving that such a 69-in-diam-driver locomotive could be operated at the same maximum speeds as the one with 79-in-diam drivers and substantiating the correctness of the method of balancing used for the converted L-2, which had taken into account the complete theoretical analysis. In this work much valuable assistance was rendered by the Timken Roller Bearing Company.

TABLE 5 SUMMARY OF MAXIMUM STRESSES; TANGENT SECTION

Locomotive	Speed range, mph	Strain-gage location on rails	Rail	No. of stresses above 15,000 psi	Five highest maximum stresses, psi: Average	Five highest maximum stresses, psi: Range
2995 L-2D (conv)	54.4 to 87.2 (9 runs)	Underneath the rail	Left (S)	64	21400	22100 to 20900
			Right(N)	17	18800	23300 to 17100
5330 J-1E	66.2 to 85.9 (5 runs)	Underneath the rail	Left (S)	34	21200	22600 to 19500
			Right(N)	16	18800	20500 to 17800
5435 J-3A	72 to 83.2 (4 runs)	Underneath the rail	Left (S)	21	24200	29100 to 20200
			Right(N)	11	19200	20800 to 18300
2995 L-2D (conv)	54.5 to 87.2 (9 runs)	Outside of head of rail	Left (S)	43	20500	23000 to 18500
			Right(N)	26	20200	23400 to 18600
5330 J-1E	66.2 to 85.9 (5 runs)	Outside of head of rail	Left (S)	33	19400	20500 to 18500
			Right(N)	20	22000	27700 to 19100
5435 J-3A	72 to 83.2 (4 runs)	Outside of head of rail	Left (S)	19	17700	18400 to 17400
			Right(N)	10	19500	22000 to 18400

The ranges of comparative stresses for the two locomotives and for one of the J-3 class tested at the same time are given in Table 5.

New L-3 Combination Passenger-and-Freight Locomotive

On the basis of the experience gained with the two converted L-2 engines and the study that had been given to the design, with the close cooperation of The American Locomotive Company, the Superheater Company, the Timken Roller Bearing Company, and others, fifty of the L-3 class were ordered and have lately been delivered, twenty five of which are arranged for operation in either passenger or freight service, while the remaining twenty five are strictly freight locomotives but having the same characteristics with respect to speed versus track structure. The combination engines are equipped with a cast-steel pilot and drop coupler, steam heat, air signal, engine-truck brake, and roller bearings on all wheels including drivers. Boosters were omitted although arrangements were made for convenient application if subsequently found desirable.

The 4-8-2 wheel arrangement was retained as it was found that for this design the required weight distribution could be secured without the use of a 4-wheel trailing truck and also because it was desired to supply the largest possible tender, particularly with reference to coal capacity, without extending the over-all length of engine and tender beyond the limits of the 100-ft turntables now in use at principal main-line terminals.

The extra large coal capacity of 43 tons was provided to increase materially the length of runs and through intensive use to obtain high monthly mileage which would be equivalent to additional locomotives.

A waterscoop of improved quick-acting design, which had recently been developed and tested, and which supplies approximately 20 per cent more water with a substantial reduction in the amount spilled, was applied.

The standard 69-in. driving wheels were retained as experience had proved this size best for high-speed main-line freight service in which the engines would be used the greater portion of the time. Provision was made, however, by increasing the driving wheel base and the over-all length, for the future application of 72-in. driving wheels, as a margin of protection for high-speed running.

For reasons beyond the scope of this paper, it was decided to use carbon steel instead of nickel steel for the boilers and, in order to conform to the desired limits of total weight and wheel loads, this necessitated using a working boiler pressure of 250 psi instead of the 275 psi originally planned. With this pressure and cylinders of $25^1/_2$ in. diam and 30 in. stroke, a rated starting tractive effort of 60,100 lb was obtained, about equal to the 60,620 lb of the L-2 class as desired.

The increase in driving wheel base permitted the use of a combustion chamber 12 in. longer than on the L-2 for increased firebox volume and greater combustion efficiency; greater gas area through an enlarged superheater was provided to increase the superheat temperature.

An improved front end, as developed by the Selkirk tests heretofore mentioned, was installed.

Particular attention was given to the proportioning of steam passages from dome to exhaust to provide free steam passage and reduce transmission losses, and the large-volume steam chest with the standard 14-in. valves was retained.

Reciprocating parts are of special lightweight design similar to those used on the two converted L-2 class, and all wheels were dynamically balanced in accordance with the theoretically correct principles established for those engines.

The following modifications were made with a resulting decrease in weight:

Cor-Ten steel main air reservoirs.

Aluminum cab, running boards, cylinder and valve casings dome and turret casings, and gage board.

High-tensile-steel drop coupler.

Lightweight magnesia block lagging.

Tubes and flues to one gage tolerance.

New design lightweight valve gear.

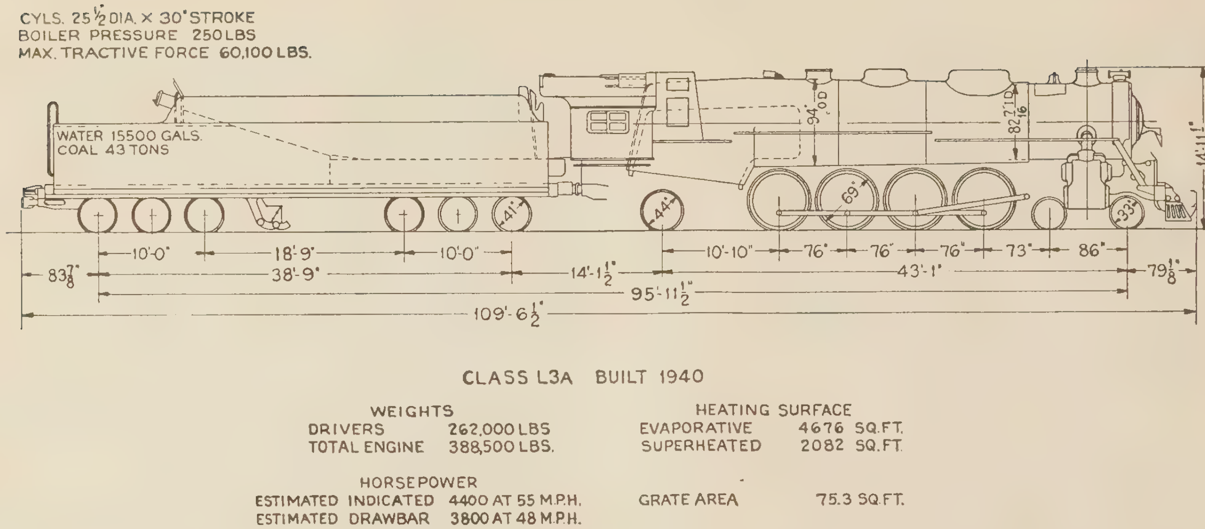


Fig. 15 Characteristics and Principal Dimensions of Latest Combination Passenger-and-Freight Locomotive, Class L-3a

Other special features incorporated were complete speed-recorder and cutoff-selection equipment, coal pusher, Alemite grease equipment for rods and other parts, roller bearings on all axles, and lateral-motion device on front and main drivers.

TABLE 6 CHARACTERISTICS OF L-3 TYPE LOCOMOTIVE COMPARED WITH L-2D FREIGHT AND L-2D CONVERTED

	L-2D Freight	L-2D Converted	L-3A Combination
Wheel base:			
Driving		18 ft 0 in.	19 ft 0 in.
Engine		42 ft 1 in.	43 ft 1 in.
Engine and tender		84 ft 8 in.	95 ft 11 1/2 in.
Weight in working order, lb			
Engine truck	59150	65400	70400
Drivers	250000	257000	262000
Trailing truck	61000	62700	56100
Total engine	370150	385100	388500
Tender loaded	313500	313500	373900
Boiler pressure, psi	225	250	250
Cylinders, diameter and stroke, in.	27 × 30	25 1/2 × 30	25 1/2 × 30
Rated tractive effort			
With booster, lb	73020	73850	...
Main engine, lb	60620	60100	60100
Factor of adhesion	4.12	4.28	4.36
Permissible operating speed, mph	60	80	80
Boiler:			
Inside diameter, first course, in.	..	82 7/16	82 7/16
Outside diameter, third course, in.	..	94	94
Tubes, number and diameter, in.	..	40–2 1/4	50–2 1/4
Flues, number and diameter, in.	..	199–3 1/2	198–3 1/2
Length of tubes and flues, ft-in.	..	20-6	20-6
Firebox, length and width, in.	..	120 1/8 × 90 1/4	120 1/8 × 90 1/4
Grate area, sq ft	..	75.3	75.3
Combustion chamber length, in.	..	51	63
Firebox volume, cu ft	..	510	538
Gas area through flues and tubes, sq ft	..	9.45	9.33
Heating surface, sq ft			
Firebox	..	354	373
Tubes and flues	..	4202	4303
Total	..	4556	4676
Superheater	..	1931	2082
Tender:			
Coal capacity, tons	..	28	43
Water capacity, gal	..	15000	15500
Size of axle journals, in.	..	6 × 11	6 1/2 × 12
Cylinder horsepower	3800 at 48 mph	4200 at 50 mph (est)	4400 at 55 mph (est)
Engine weight per horsepower, lb	97	92	88 (est)

The estimated drawbar pull and horsepower versus speed for the L-3 are shown by the curves included in Fig. 8. Capacity and performance tests for the L-3 are now in progress and the characteristics shown are believed to be conservative.[5] The principal dimensions and the proportions are shown in Fig. 15.

[5] "New York Central Buys All-Round Road Locomotives," *Railway Age*, vol. 109, 1940, pp. 856–861, 864.

"New York Central Buys 50 4-8-2 Type Locomotives," *Railway Mechanical Engineer*, January, 1941, pp. 1–8, 21.

TABLE 7 SUMMARY OF PRINCIPAL WEIGHT AND POWER CHARACTERISTICS FOR LOCOMOTIVE DESIGNS DISCUSSED

Class	Type	Last built	Locomotive weight, lb	Maximum horsepower and speed at which attained, mph: Cylinder	Maximum horsepower and speed at which attained, mph: Drawbar	Weight per horsepower, lb: Cylinder	Weight per horsepower, lb: Drawbar
K-80	4–6–2	1912	252500	1700–39	1430–35	149	177
K-2	4–6–2	1910	273000	2000–45	1655–40	137	165
K-3Q	4–6–2	1923	295500	2100–45	1720–40	141	172
K-3R	4–6–2	1925	278000	2140–45	1750–40	130	159
K-5	4–6–2	1926	302000	3200–54	2530–45	94	119
J-1a No. 5200	4–6–4	1927	343000	3900–67	3300–58	88	104
J-1E	4–6–4	1931	358600	3900–67	3240–58	92	111
J-3	4–6–4	1937	360000	4725–75	3880–65	76	93
Converted L-2	4–8–2	1930	385100	4200–50	3640–43	92	106
L-3	4–8–2	1940	388500	4400–55	3800–48	88	102

For convenient reference, some of the characteristics, as compared with the L-2D converted class and the L-2 freight engine, are given in Table 6, the data for the L-2 freight only being shown where different from the converted L-2.

Present Thoughts on Trends of Steam-Locomotive-Design Improvement for the Near Future

While this paper is confined to the subject of the conventional steam passenger locomotive, the author is fully cognizant of the rapid strides being made by other forms of motive power, their possibilities, advantages, and growing importance to the railroads for certain classes of service.

This may be illustrated by stating that on the New York Central, 127 Diesel-electric locomotives are used in intensive daily service. As early as 1924, a 60-ton 300-hp Diesel-electric locomotive was operated in switcher and puller service in New York City territory with favorable results, followed in 1928 by a road freight and in 1929 by a road passenger locomotive. The first straight electric was introduced in 1904, and there are now 168 of various types and capacities in use on the System. Within the last 6 years, limited operating experience has been obtained with a 5000-hp experimental turboelectric locomotive and a 3600-hp Diesel-electric, both designed for high-speed main-line service, and a 5400-hp Diesel-electric freight locomotive.

Future development of the steam locomotive in some radically new form, such as the steam-turbine condensing or combustion type, as recently proposed, should show a substantial increase in thermal efficiency but, until the stage has been reached where such units of proved dependability in daily operation can be pro-

duced of moderate size, weight, and cost, it is the author's belief that basic lines of development should be continued by taking advantage of the possibilities for further betterment of the conventional reciprocating design without radical changes in the type of boiler or resorting to the mechanical complication of multiple expansion of steam. It should be possible now to produce a highly serviceable two-cylinder single-expansion locomotive of the 4-8-4 type at a weight per indicated horsepower closely approaching that represented by the 4-6-4 class J-3A described in this paper, capable of delivering 6000 cylinder hp when required.

Such a design should include the largest practicable superheater, with ample firebox volume and grate area, carefully proportioned steam passages from boiler to exhaust, and a working steam pressure probably up to 300 psi.

Roller bearings on all locomotive and tender axle journals and to a lesser degree in rods and motion work have resulted in increased serviceability because of freedom from heating failures. Their extended utilization should receive careful consideration.

For the future extension of steam-locomotive productive capacity, design study leading to a better proportioned and more efficient boiler is proposed. The development of a suitable drier arrangement to provide high-quality steam, taken directly from the boiler barrel, would permit elimination of the steam dome with corresponding possibilities within given weight or clearance limitations for increased diameter of barrel with improved tube and flue layouts and larger gas areas and superheater, additional firebox depth and volume, and more nearly level grates without restricting the highly important features of adequate ash-pan capacity and arrangement necessary for long locomotive runs.

Design studies and performance experiments are now in progress to improve the poppet-valve arrangement of steam distribution and these efforts may result in making available for practical use the better cylinder performance in relation to power output and efficiency inherent therein, without prohibitive increase in the size and weight of the boiler.

Other interesting experiments now in operation include locomotives having four simple cylinders and two separate sets of running gear or combined within a single rigid wheel base, with which improved wheel loadings and rail effects should be obtained, together with lower dynamic forces in machinery and running-gear parts, as well as other advantages.

Ability to extend the length of locomotive runs in either freight or passenger service without stops for fuel depends to a great extent upon the size of the tender. This, in turn, may be limited by possible restrictions on over-all length. One leading western railroad now has in service back of a considerable number of modern design steam locomotives a new arrangement of tender running gear and underframe which possesses possibilities of materially increased tender capacity within given dimensional restrictions.

Discussion

T. V. BUCKWALTER.[6] This paper reviews the detailed development of the six-coupled locomotive, starting with the New York Central class K-80 in 1904, to its ultimate development, culminating in the Hudson type class J-3A in 1937. The improvement in locomotives from year to year is gradual and does not afford an outstanding contrast when comparing the 1904 locomotive with the 1937 locomotive. The improvement of from 1430 to 3880 drawbar horsepower has been made in the short span of 33 years, the power development of 1937 being 2.71-fold that of 1904, or an annual increase of 8.2 per cent.

This development was made in two stages in so far as the New York Central System is concerned, the first being the Hudson type class J-1E developed in 1927, developing 3300 hp, an increase of 30 per cent over that of its predecessor, the K-5 locomotive, built in 1926, and the second step, the Hudson J-3A in 1937.

The Hudson locomotive served the New York Central System so well that a total of 225 were built, replacing the older "K" type Pacific locomotives on main-line service.

The Hudson type was further improved in 1937, increasing capacity in drawbar horsepower to 3880, or 17 per cent; this in itself being an outstanding improvement, considering that the weight on the drivers of the J-3A is only 11,000 lb more than that of the J-1E, while the actual engine is only 1400 lb additional weight. The use of lightweight reciprocating parts, effecting a reduction of more than 50 per cent in weight, afforded an opportunity to redistribute the counterbalance, the typical practice being to distribute $66^1/_3$ to 75 per cent of this reciprocating-weight reduction to lightening the counterbalance and, therefore, the dynamic augment on the rail, and utilizing the balance to reduce nosing. Together these have an important influence on improved riding qualities of the locomotive and also tend to reduce rail reaction. As the author mentions, the Hudson engines have operated 200,000,000 miles without a single case of rail damage attributable to this class of locomotive.

A further outstanding development, since the origin of the Pacific engine, is the increase of speed from 35 mph to 65 mph, at which maximum horsepower is developed; this being an increase of 85 per cent. At the same time the weight reduction per draw-

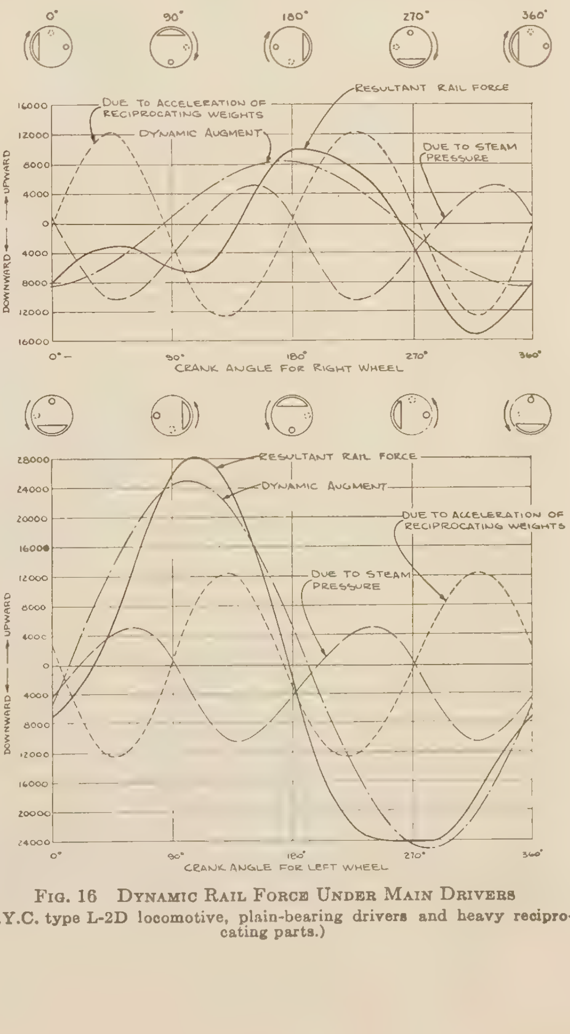
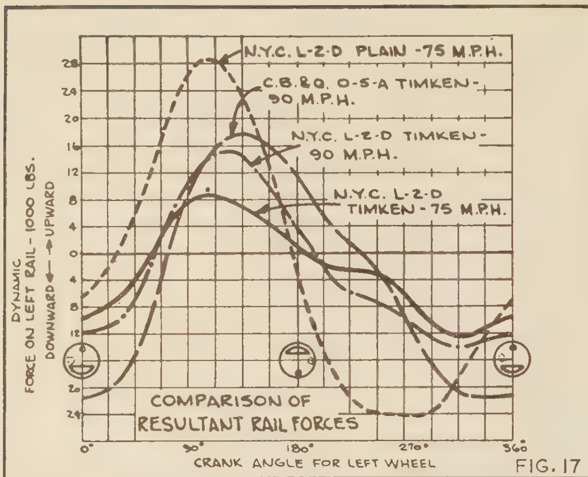
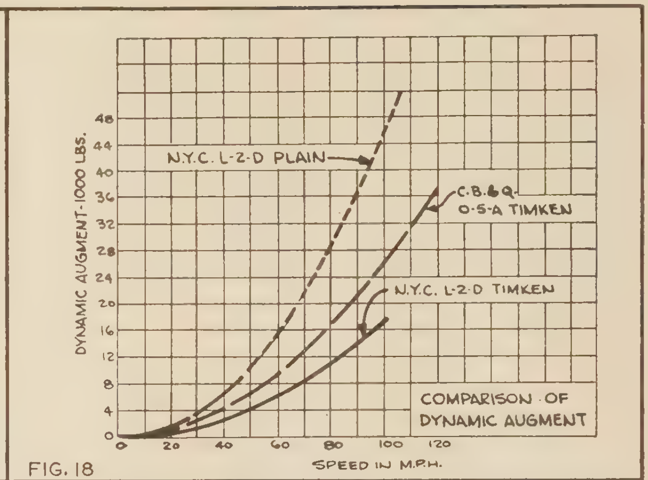
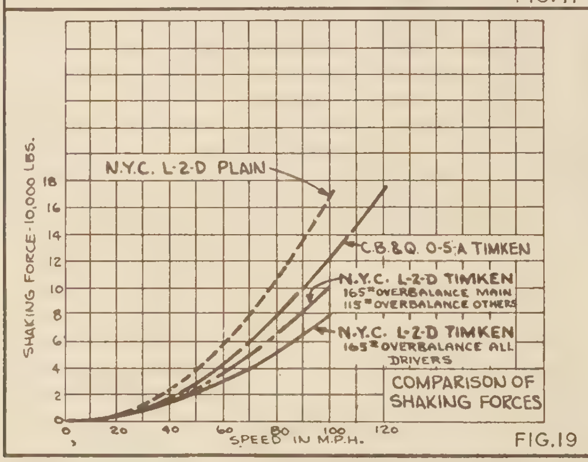


FIG. 16 DYNAMIC RAIL FORCE UNDER MAIN DRIVERS
(N.Y.C. type L-2D locomotive, plain-bearing drivers and heavy reciprocating parts.)

[6] Vice-President, Timken Roller Bearing Company, Canton, Ohio. Mem. A.S.M.E.







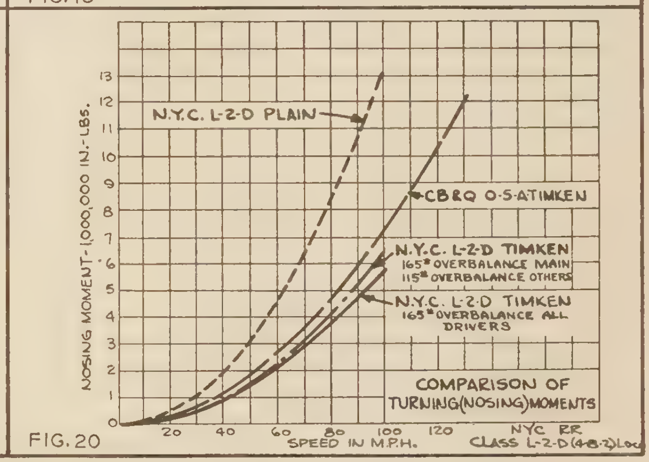


Fig. 17 Comparison of Resultant Rail Forces

Fig. 18 Comparison of Dynamic Augments

Fig. 19 Comparison of Shaking Forces

Fig. 20 Comparison of Turning, or Nosing, Moments

bar horsepower from 177 lb to 93 lb, or 48 per cent, is noteworthy.

Probably the major improvement in steam passenger locomotives is the increase in capacity for work, as measured in ton-miles per month. The average of the fifty J-3 locomotives for the month of December, 1940, was 11,689 miles each, giving effect to engines in shop and engines partially employed during the month. Ten of these locomotives actually made over 16,000 miles in that month. It is doubtful whether the typical passenger locomotive prior to 1925 averaged more than one third of that figure and handled trains, giving effect to the air-conditioning drag about one half as heavy. This would indicate that the modern Hudson locomotive has 6 times the capacity for work, as compared with the locomotive of only 15 years ago. This is a noteworthy development and affords a further indication that the replacement of the steam passenger locomotive by other forms of motive power is still a long distance in the future.

The New York Central received delivery of fifty class L-3 Mohawk locomotives during 1940. This followed the experimental conversion in 1939 of two class L-2D Mohawk, type 4-8-2 freight locomotives. Much thought was given to the conversion locomotive No. 2998. The reciprocating weight was reduced from 2143 to 1239 lb, a reduction of 904 lb. The overbalance in the plane of the rail on the left main driver was reduced from 441 lb to 171 lb and a corresponding dynamic augment at diameter speed from 21,200 lb to 8220 lb, a reduction of 12,980 lb, or 158 per cent of the remaining dynamic augment.

The rail reaction on the main driver at 75 mph was reduced from 28,100 lb to 8200 lb, upward against the spring rigging, and from 24,000 lb to 12,200 lb, downward against the rail. The nosing moment was reduced from 8,400,000 in-lb to 4,000,000 in-lb at 80 mph, and the fore-and-aft shaking force from 108,000 lb to 64,000 lb. These figures are reflected in the relatively low rail stresses at speeds up to 87 mph, as indicated in Table 5 of the paper.

Curves of dynamic rail force under the main drivers for the L-2D Mohawk are shown in Fig. 16. Comparisons of resultant rail forces, dynamic augments, shaking forces, and turning moments of this type locomotive with others at various speeds are given in Figs. 17 to 20, inclusive.

The Mohawk engine, having 69-in. drivers, balanced as described, ranks with the best of the results obtained from the J-1E and J-3A Hudson type locomotives at high speeds measured in lower stress in rails and good riding qualities.

The experience with 25 of the L-3's, equipped for passenger-

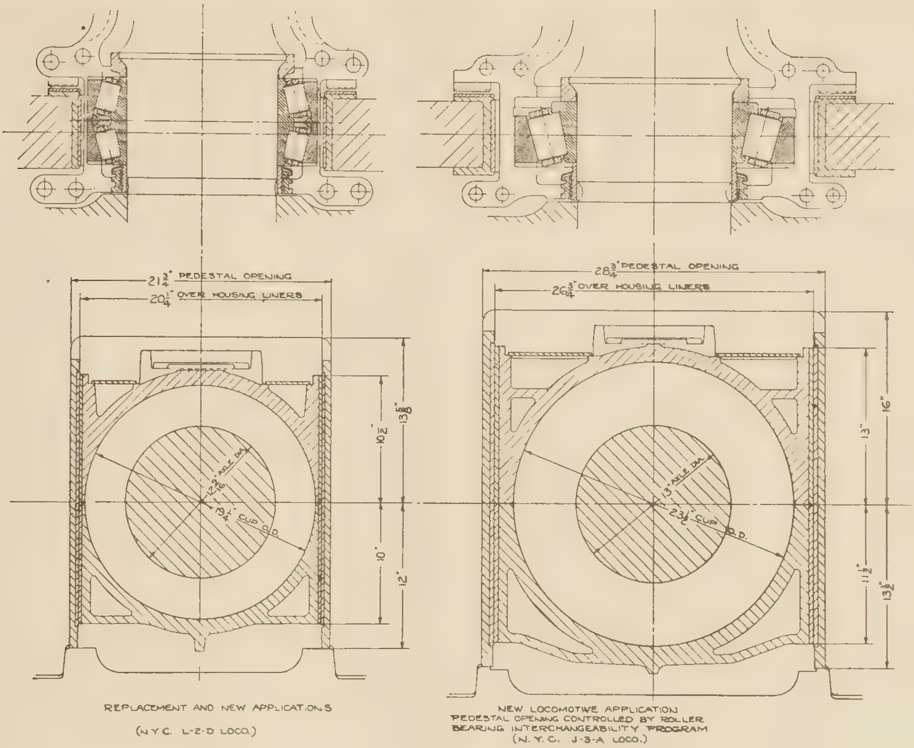


Fig. 21

train operation, proves that the new Mohawks can be used interchangeably in passenger and freight service. This equipment defers the increase in Hudson locomotive inventory for the handling of peak loads in midwinter and midsummer, and the Labor Day holidays. At the same time, these locomotives are available for freight-traffic peaks developing in the early fall. Notwithstanding the general improvement in passenger service, this locomotive has also been improved in its capacity to handle freight traffic.

Table 7 of the paper, showing the capabilities of this locomotive for high-speed operation as ranking with the best passenger locomotives, together with its greatly increased drawbar capacity at low and medium speeds, indicates that this locomotive is capable of handling heavy passenger and express trains in congested traffic at a higher average rate of speed than any other locomotive available. It is believed to be the outstanding locomotive achievement in recent years.

The New York Central J-3A (Hudson) and L-3A (Mohawk) locomotives afford an additional interesting comparison, shown in Fig. 21, entitled "main-pedestal-opening comparison." The J-3A driver-bearing application, shown at the right in Fig. 21, is based on interchangeability with other types of roller bearings. The pedestal opening of $28^3/_4$ in. and pedestal clearance of 16 in. above the center line of the axle are derived from the space requirements of other roller bearings.

The L-2D and L-3A pedestal opening is based on the space requirements of the taper roller bearing of the double-row type. The pedestal opening is $21^3/_4$ in. and the pedestal clearance is $13^5/_8$ in. which interchange exactly with the plain-bearing requirements. In addition, there are outstanding reductions in unsprung weight and provision of more space for spring rigging with the advantages of an equal degree of reliability and greater economy in the plain-bearing interchangeable layout. Most current locomotive construction is based on this interchangeable layout.

R. M. Ostermann.[7] This paper discloses a very creditable engineering achievement. It is remarkable how greatly the sustained capacity of the New York Central J class locomotives was increased by attention to details and by refinement in design. In the section "Thermal Efficiency at Tender Drawbar Referred to Fuel," the author explains the circumstances which led him to compromise between that thermal efficiency and practical operating advantages, and he thus exhibits the very logical view of a railroad operating man.

However, the engineer sitting on the sidelines wonders whether steam-locomotive designers will not eventually be forced to far-reaching modifications of the design of steam locomotives because of the pressure of the competition which the conventional steam locomotive is experiencing from the Diesel-engine-powered locomotive. This type of motive power has great advantages of thermal efficiency. Steam-locomotive designers may be forced, in self-defense, in their future designs, to embody some of the principles, the application of which has produced such eminent progress in the economy of stationary power plants within the last 15 years. For instance, it is seen that the J-3 engine, at its best, works with a heat drop per pound of steam of about 160 Btu. It is not at all unreasonable to expect that steam locomotives may someday be worked with nearly double that heat drop, in an entirely practical manner.

As the author points out it is perfectly true that, in railroad locomotives, we are confronted with acute limitations of weight, height, width, and length of structure, but it seems to the writer that, just because of them, every physical means should be tried

[7] Vice-President, Western Territory, The Superheater Company, Chicago, Ill. Mem. A.S.M.E.

in order to get a maximum of capacity into a given cubic space. The writer feels that men such as the author who have been able to refine the design of conventional steam locomotives to such a remarkable degree, by painstaking engineering analysis, will also be able to make entirely new forms of steam locomotives practicable.

James Partington.[8] In this paper we have a very complete picture of the development of main-line steam locomotives for passenger service. These modern locomotives show wonderful sustained horsepower output and are making a yearly mileage of well over 200,000.

To keep within the railroad company's weight limits on the L-3 locomotives, the cabs, runboards, and dome casing are made of aluminum. On the J-3, all of the important casings, dome casing, cylinder-head casing, etc., were aluminum, and the cabs were aluminum on forty engines; ten engines were Cor-Ten steel. Here we have a situation where the saving of weight was urgently necessary and the use of expensive material was resorted to for a weight saving of about 2000 lb per locomotive. Twice this saving could be obtained if the boilers of these locomotives were of welded construction. Therefore, the writer would like to discuss briefly the welded boiler for locomotives.

Soon after the rules for fusion welding were adopted by the A.S.M.E. Boiler Code Committee, they were placed in the Code for Power Boilers. The use of these rules proved them to be safe and satisfactory, and after several years' experience the Code Committee suggested that they be incorporated into the Code for Boilers for Locomotives.

After several conferences with the I.C.C. Bureau of Locomotive Inspection, and largely through the efforts of the late L. F. Loree, then president of the Delaware & Hudson Railroad, permission was granted to that company to build and operate a locomotive with a welded boiler. This boiler was built by the American Locomotive Company to meet the requirements of the A.S.M.E. Power Boiler Code, and the locomotive was placed in freight service on the D.&H. in the fall of 1937. This welded boiler will soon have a service record of 4 years. During this time thorough inspections made frequently have shown that it has a performance record of 100 per cent.

When the Interstate Commerce Commission gave its permission for the operation of this locomotive with a welded boiler, it was in response to a request sponsored by many of the leading railroads that an experimental installation be allowed. In granting this request it was stipulated that no additional welded boilers would be permitted pending a test period of 5 years. For the operation of additional locomotives with welded boilers, the railroads will have to secure authorization from the I.C.C. through a procedure similar to that used by the D.&H. Company for this first welded boiler.

The advantages of the welded locomotive-type boiler are that there are no rivets; no overlaps; no obstructions inside or outside; no joint repairs or failures; lower upkeep expense; higher efficiency; lighter weight; easier handling; quicker washing; neater appearance. Finally, this form of construction eliminates caustic embrittlement, which has caused rivet failures and cracked sheets in the region of both circumferential and longitudinal seams of riveted boilers. In a number of cases this has made expensive repairs necessary.

Welding has supplanted riveting for high-pressure stationary boilers and for nearly all pressure-vessel construction.

There are several hundred locomotive-type welded boilers in use for power purposes in the United States, and these boilers are operating at pressures up to 350 psi.

[8] Manager, Engineering Department, American Locomotive Company, New York, N. Y. Mem. A.S.M.E.

Will more welded boilers for locomotives be built?

This will depend upon the attitude of the railroads and the decision of the Interstate Commerce Commission.

The locomotive builders are ready to give full assistance and cooperation in this development

F. E. Russell.[9] Our experience with the development of high-speed steam locomotives agrees closely with that of the author, in that we have been able to increase the efficiency and output of the steam locomotive by intensive development and improvement in design. By providing ample firebox volume and gas area through tubes and flues, it is possible to increase the evaporation considerably without excessive losses in combustion and absorption efficiency, as is shown by the performance curves in the author's paper. In this regard, on the Southern Pacific, we are more fortunate, in that the combustion efficiency of the locomotive boiler with oil fuel is high, and there is little falling off in over-all boiler efficiency as the firing rate is increased; and high outputs can be obtained with a moderate draft and back pressure.

Steam-flow conditions are extremely important if the maximum potential output of a steam locomotive is to be obtained in practice. The dry pipe, superheater header, and superheater units should be of ample area, and the flow should be as direct and "streamlined" as the design will permit. The writer considers the design of the steam and exhaust passages in the cylinder especially important. Much of the increase in horsepower per unit volume in our more recent locomotives is due to careful cylinder design, with our insistence on definite steam, exhaust, and valve bushing area in relation to piston size. In this respect the increase in boiler pressure, with subsequent decrease in cylinder size for a given power, has been a great advantage, as it is possible to provide greater area in the passages per unit volume of steam handled.

Prior to 1927, apparently little attention was paid to increasing the area of exhaust passages in the cylinders in proportion to the cylinder diameter, resulting in high back pressure and low power output at high speeds. During that year we made an analysis of the proportion of exhaust passages, as related to piston area, and since then all our locomotives have been built with improved exhaust passages, resulting in increased horsepower output and reduced back pressure and decreased fuel consumption.

In 1920, we made some dynamometer tests on 2-10-2 type superheater locomotives and determined that for superheated locomotives we should get away from saturated-steam practice. It was also decided that, in order to obtain maximum fuel economy, the cylinder diameter should be reduced and the stroke increased over the proportions normally used at that time. In 1921, when we purchased our first heavy Pacifics for service between Sparks, Nevada, and Ogden, Utah, handling heavy transcontinental trains, we went to a 30-in. stroke with a 25-in. cylinder, using 73-in. drivers. These locomotives were so satisfactory in passenger service, not only in high-speed hauling capacity but also in their ability to start heavy trains without taking slack, that we have continued the practice of using a relatively long stroke. Our latest 4-8-4 type locomotives have $25^1/_2$-in. × 32-in. cylinders with 80-in. drivers. We find these proportions very satisfactory for starting heavy trains and for high-speed full-power operation.

Given a boiler designed for high output and high superheat, and steam and exhaust passages capable of handling the steam required for high power output, the remaining problem is that of utilizing this steam with the greatest efficiency in the cylinders. In this regard, the present form of piston valve with interconnected timing of events is certainly not ideal. The writer con-

[9] Mechanical Engineer, Southern Pacific Company, San Francisco, Calif. Mem. A.S.M.E.

fidently looks forward to the perfection of some form of valve with independent timing, to obtain more adequate valve openings at high speeds and short cutoffs, together with a reduction in the present distortion of exhaust events. In this connection, the poppet-valve gear mentioned in the paper is of great interest, and is believed to be very promising.

As an example of what a modern high-power steam locomotive can do, the writer has analyzed some of the performance records of our latest 4-8-4 type passenger locomotives purchased in 1940. These locomotives have cylinders $25^1/_2$ in. × 32 in., with 80-in. drivers, and carry 300-psi boiler pressure. They are equipped with type E superheaters, Worthington SA feedwater heaters, and have oil lubrication on all axle bearings. These locomotives are used to handle the streamline "Daylight" trains between San Francisco and Los Angeles, which are frequently as heavy as 16 cars, weighing 924 tons loaded. On these trains, the portion of the run that requires the highest power output is eastbound from Camarillo to Santa Susanna, Calif., a distance of 20.9 miles, with an average opposing grade over the entire district of 0.75 per cent. Between these points the schedule speed is 62.7 mph. Examining the speed-record tapes, the writer finds that, on one 7-mile continuous stretch of compensated 1 per cent grade in this territory, the maximum speed maintained steadily is 55 mph with the 16-car 924-ton train, which performance requires a calculated drawbar horsepower on the level of 4750; the equivalent cylinder horsepower is estimated at 5400. On other occasions, these locomotives have handled the "Daylight" with 13 cars, weighing 729 tons, on the 2.2 per cent Cuesta grade, making speeds of 28 mph without a helper.

On our Salt Lake Division, a slightly older type of 4-8-4 locomotive is handling trains of 20 cars, weighing approximately 1400 tons, on fast schedules.

The writer is in complete agreement with the author that research and design have greatly improved the high-speed steam passenger locomotive, and that continued development and testing will result in future locomotives of higher output and efficiency than yet obtained.

C. J. Surdy.[10] We are fully aware that present-day traffic demands result in a radical change in operating methods. In many instances this traffic is seasonal and, during such periods, there is a heavy demand for high-speed power in both passenger and freight service. Few, if any, roads can afford to hold in reserve power to meet such seasonal demands, therefore, it is quite evident that the solution of this problem is the dual-service locomotive, described so well in this paper.

Of more than usual interest is that portion of the paper in which the author presents his thoughts on trends of steam-locomotive design and improvement for the future. With the experience on the New York Central, including the experimental use of a 5400-hp Diesel-electric locomotive and a 5000-hp turboelectric locomotive, the author's endorsement of the conventional reciprocating-design locomotive should be regarded by motive-power designers as a challenge. Apparently, his choice of motive power is made on the basis that nothing better has yet been offered, when due consideration is given to all factors.

There is much more to consider in the selection of motive power than the known advantages which are derived from more expensive power, such as Diesel-electrics or turboelectrics of the experimental type proposed in recent years. Practical railroad men must also give full consideration to the economic forces which, in a considerable measure, determine whether or not coal must be used as a source of power. They must also take into account the effect which their use of certain kinds of coal may have on the consumer market which their railroad serves. Obviously, to take off the market a coal that is desired by the consumer only results in loss of the haul to the railroad. In 1939 bituminous coal accounted for one sixth of the total railroad freight revenue in the United States. From this, it appears the railroads need the coal freight haul, while the coal-mine operators apparently need the railroad fuel business to stabilize their output.

On the premise that many railroads will continue to regard coal as a primary fuel for their motive power, it appears that sufficient incentive exists for locomotive designers to give more thought to producing a coal-burning motive-power unit with the characteristics of the Diesel-electric. Some work of a preliminary nature has been done in this respect, but apparently the development is not complete.

It is not within the scope of this discussion to promote a new system of motive power, but this paper, which takes us step by step through the development of steam locomotives on the New York Central, indicates that, as greater demands are made for sustained power output of steam locomotives, more study will have to be given to some arrangement for increasing the diameter of the boiler barrel which will result in improved tube and flue layouts with larger gas areas and superheater. Since the diameter of the boiler cannot in any event, with steam-locomotive-design trends leading toward larger drivers, be increased greatly over present dimension, the thought of other systems of motive power naturally arises.

With the wide experience gained through the use of the locomotive boiler, of simple construction and high steam-generating capacity, a logical starting point for development of a turboelectric locomotive is present. The boiler of a New York Central Hudson-type locomotive can generate a maximum of about 100,000 lb of steam per hr. By the use of a water-tube firebox, it should be possible to generate steam at a pressure in the neighborhood of 700 psi. This capacity would exceed the steam requirements of two or even three 2500-hp turbines, so that a 7500-hp turboelectric locomotive, using coal as a primary source of fuel in a boiler and firebox somewhat along present conventional lines, is within the realm of possibility.

Author's Closure

The author is grateful for the privilege of submitting brief comments on the valuable and helpful discussions contributed to the subject matter of his paper.

Mr. Buckwalter's discussion has been reviewed with much interest and appreciation. The material presented is consistent with our experience and records. The information given in terms of "increase in capacity for work," as pertaining to the modern steam passenger locomotive, provides a practical illustration of progress in this field.

Mr. Buckwalter refers to the increase of 17 per cent in drawbar horsepower obtained in the Hudson type locomotives with an increase of only 11,000 lb on drivers and 1400 lb in total weight. It should be pointed out that the increase of 11,000 lb on drivers was arranged deliberately to produce a higher factor of adhesion for this extremely high-powered locomotive, and was practicable because of the reduced rail reactions secured by the methods indicated. If considered desirable, the small increase in total weight could have been distributed to leading and trailing trucks by proper equalization, with weight on drivers held to that of the previous Hudsons.

Later in his discussion Mr. Buckwalter refers to the latest class L-3, equipped for passenger-train operation, the estimated power characteristics of which were shown in Table 7 of the author's paper.

Preliminary results of the tests of this class conducted under

[10] Assistant to General Manager, The Standard Stoker Company, Inc., New York, N. Y.

regular road service conditions of operation may now be reported as follows

Maximum cylinder horsepower...........	5120 at 67 mph
Maximum drawbar horsepower..........	4320 at 62 mph
Engine weight per cylinder horsepower...	76 lb
Engine weight per drawbar horsepower...	90 lb
Drawbar pull at 85 mph................	17000 lb

The final report has not as yet been completed, and minor corrections may be required after all dynamometer records have been thoroughly checked.

Although Mr. Ostermann has commented favorably on the increase in sustained capacity on the New York Central class J locomotives through attention to detail and by refinement in design, he may have gained the impression from the section on "Thermal Efficiency at Tender Drawbar Referred to Fuel," that the treatment of this subject was intended to explain the circumstances which led to a compromise between thermal efficiency and practical operating advantages in the design of the Hudson type class J-3. The section on thermal efficiency was included for reasons given in the opening paragraphs, inasmuch as the paper was addressed to the subject of modern steam passenger locomotives for handling heavy high-speed passenger traffic at the present time. It is not the author's belief that it will be necessary to force steam-locomotive designers to modifications in design, because what is constantly sought by progressive railroads, locomotive builders, and manufacturers engaged in related activities, is the most efficient and economical form of motive power which may be produced to meet the changing needs of the railroad industry.

Under the concluding heading, "Present Thoughts on Trends of Steam-Locomotive-Design Improvement for the Near Future," attention has been directed to these circumstances with a view to stimulating interest and efforts of all concerned in the direction of increasing the thermal efficiency of motive-power units. However, to insure progress in rail transportation, it is necessary before introducing radically different designs in quantities sufficient to permit handling some reasonable proportion of the total traffic that such units be of proved dependability and of practicable size, weight, and cost. It is the author's belief that any contribution along these lines, from those skilled and experienced in the art, which appears to have reasonable prospect of success, would immediately receive due attention and consideration by all concerned.

Mr. Partington's discussion of the welded boiler for locomotives is pertinent and timely. He has supplied a concise outline of the numerous advantages to be gained through this form of locomotive-boiler construction, which has become widely used in the fabrication of boilers for stationary plants, marine and other services, and for various types of pressure vessels.

It may be said that this subject is now under active consideration through official channels of the Association of American Railroads in cooperation with the locomotive builders. It is hoped that, through negotiations with the Interstate Commerce Commission, additional boilers for locomotive use may be constructed by the fusion-welding process and placed in service within a reasonable time.

The author is gratified to note that Mr. Russell's experience agrees with his own on the matter of increased efficiency and capacity of the steam locomotive, obtained by extensive development and improvement in design, with particular reference to steam-flow conditions, and shares his belief that continued development and testing will result in future steam locomotives of still higher output and efficiency than any yet obtained. Further work of this character is being actively prosecuted by the New York Central, and the hope is expressed that a similar procedure will be carried on throughout the rail-transportation industry.

The interest expressed by Mr. Surdy and the specific ideas he has offered in reference to the thoughts presented on trends of steam-locomotive design and improvement for the future have been noted with appreciation. As indicated under this section of the paper, extensive experience has been obtained on the New York Central with certain types of Diesel-electric locomotives, and the results have been fully up to expectations. At the same time, limited experience has been gained with other experimental locomotives, designed for road service, and continued progress is anticipated in the design of steam, Diesel, and other forms of motive power. As this progress continues, supplemented by service and economic experience, a wider choice of motive-power types should be made available, subject to an analysis of the different conditions to be met.

Francis-Turbine Installations of the Norris and Hiwassee Projects

By GEORGE R. RICH[1] AND J. F. ROBERTS,[2] KNOXVILLE, TENN.

The Hiwassee turbine, having a rated output of 80,000 hp at 190-ft head, is believed to be the highest-powered Francis wheel in the eastern United States. The Norris turbines are identical in physical size with the Hiwassee unit and have a rated output of 66,000 hp at 165-ft head. In this paper, considerations affecting the selection of these machines are discussed, and principal features of the design and acceptance testing are described.

THE Tennessee Valley Authority is a Federal corporation invested with statutory power to develop the Tennessee River in the combined interests of navigation, flood control, and electric-power generation. The ultimate plan of development will provide a channel for 9-ft navigation for a distance of 650 miles from the confluence of the Tennessee with the Ohio at Paducah, Ky., to the headwaters at Knoxville. The flood-control storage afforded by the main river and tributary storage reservoirs, about 10,000,000 acre-ft, will be sufficient to reduce flood crests on the Mississippi River about 2 ft and possibly 3 ft at Cairo, Ill. The ultimate installed hydroelectric capacity will be about 2,000,000 kw. As part of the National Defense Program, there is now under construction a modern steam plant of 120,000-kw capacity for the purpose of priming high-grade secondary hydro energy during periods of water deficiency. This station, together with other properties obtained by acquisition, will afford a total steam capacity of about 300,000 kw, giving a high degree of flexibility to the system, which is favorably located in close proximity to an extensive supply of high-grade steam coal. Fig. 1 and Table 1 show the geographic location and physical features of the various water-control projects.

The Norris and Hiwassee projects are essentially storage developments for operation at low capacity factor on the non-navigable tributary streams, and have the primary function of impounding headwater floods and augmenting the flow of the main river in the interests of navigation and power during the dry season. Long-term records establish the low-flow period from September 15 to December 1, in an average year and from July 15 to January 15, in an extremely dry year. Both long-term records and modern analyses of storm movements indicate that floods of major magnitude will occur only in the interval from December 15 to April 15. The cycle of operation of the storage reservoirs is predicated upon this hydrologic pattern. Under such rational advance planning, the requirements of navigation, flood control, and power generation are not at all incompatible, and very satisfactory operating results have been obtained.

From Table 2 it will be noted that the Norris project has a drainage area of 2950 sq miles and an average annual runoff of 4600 cfs. There are two 66,000-hp turbines which operate at heads varying from 194 ft to 129 ft. The storage available between elevations 955 and 1020, amounting to 1,500,000 acre-ft, is sufficient to provide an average daily discharge of 6000 cfs, exclusive of normal inflow, for a period of about 4 months. The Hiwassee project has a drainage area of 966 sq miles and an average annual runoff of 2150 cfs. While provision has been made for an ultimate installation of two units, only a single unit has been installed initially. This turbine operates at heads varying from 254 ft to 143 ft and is rated 80,000 hp at 190 ft head. The available storage is 365,000 acre-ft, which is sufficient to supply an average discharge of 3200 cfs, exclusive of normal inflow, for a period of about 2 months.

Reservoir Operation

The present schedule of reservoir operation is as follows: At the end of the flood period, April 15, the restriction that part of the reservoir be reserved for flood storage is lifted, and the remainder may be filled to the normal level as rapidly as stream flow permits. In a normal year, no release from storage would be required until September 15, although in a dry year, such as 1925, low flow in the lower river might require release of stored water by July 15, or on rare occasions, even during June. During the period of storing water, one unit at Norris and the unit at Hiwassee are operated as synchronous condensers with the guide vanes fully closed, but with the scroll case full of water, and with the runner and the draft tube vented to atmospheric pressure, and drawing about 2.75 per cent of normal generator rating from the transmission system. Under these conditions, the governor is so adjusted that, in case the frequency drops approximately $^1/_4$ cycle, the guide vanes open up and the turbine picks up load. During storms or other interruptions, the Norris plant has picked up as much as 50,000 kw in 8 sec. When normal conditions have been restored, the tributary plants are again motored.

Since such occurrences are infrequent and of short duration, the amount of water used is negligible, but the emergency value of such units to the generating system as a whole is considerable. While motoring, one of the Norris units consumes about 10 cfs of water, this consisting of leakage through the guide vanes and the amount required to lubricate the runner seals and stuffing box, and for cooling of the thrust bearing and the generator.

If the reservoirs reach normal full level before additional water is required for either navigation or power in the lower river, then the units are operated part of the time as generators so as just to utilize the inflow and hold the reservoirs full. Since the load on the Authority's system has been such as to require steam operation for a considerable part of each year, part-time operation of the storage plants results in a material saving of coal.

As the dry season approaches, the release from storage is gradually increased so as to maintain a steady flow in the lower river between Pickwick Dam and Kentucky Dam, where navigation is still definitely dependent upon stream flow to maintain an interim depth of 6 ft for navigation. When the Kentucky Dam is completed, probably in 1945, this stream flow will not be required for navigation on the lower Tennessee River and the storage may be released in the manner most beneficial for power generation in the Tennessee River plants and for improving

[1] Assistant Chief Design Engineer, Tennessee Valley Authority. Mem. A.S.M.E.

[2] Principal Mechanical Engineer, Tennessee Valley Authority. Mem. A.S.M.E.

Contributed by the Hydraulic Division and presented at the Spring Meeting, Atlanta, Ga., March 31–April 3, 1941, of The American Society of Mechanical Engineers.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

TABLE 1 PROJECT FEATURES

	Navigation		Reservoir						Power					
Project	Size of Lock Chamber (feet)	Maximum Lift of Lock (feet)	Area at Top of Gates (acres)	Volume at Top of Gates (acre-feet)	Controlled Flood Storage (acre-feet)	Length of Spillway (feet)	Spillway Capacity (second-feet)	Backwater Length (miles)	Rated Head (feet)	Head for Best Efficiency (feet)	Present[4] Plant Capacity (kw)	Ultimate[4] Plant Capacity (kw)	Effective Capacity During High Flow - 1926-1927 Flood (kw)	Type of Turbines
Kentucky[1]	110x600	73	256,000	6,100,000	4,570,000	960	1,100,000	184.4	48	51	--	160,000	74,000	Kaplan
Pickwick Landing	110x600	63	46,800	1,091,000	418,000	880	670,000	52.7	43	56	72,000	216,000	36,000	Kaplan
Wilson	60x300[2]	90	16,200	600,000	--	2,212	629,000	15.5	95 & 92	95 & 92	184,000	444,000	419,000	Francis
Wheeler	60x360	53	68,300	1,150,000	429,000	2,400	687,000	74.1	48	48	64,800	259,200	245,000	Propeller
Guntersville	60x360	45	70,700	1,019,000	282,000	720	625,000	82.1	36	36	72,900	97,200	69,000	Kaplan
Hales Bar	60x267	37	5,800	126,000	--	1,200	--	39.8	35	36	50,500	50,500	20,000	Francis
Chickamauga	60x360	56	37,200	655,000	377,000	720	600,000	59.0	36	48	81,000	108,000	67,000	Kaplan
Watts Bar[1]	60x360	70	41,500	1,132,000	370,000	800	550,000	72.4	52	52	90,000	150,000	150,000	Kaplan
Fort Loudoun[1]	60x360	80	14,900	365,500	105,000	800	550,000	55.0	65	70	--	96,000	64,000	Kaplan
Norris	--	--	40,160	2,567,000	2,020,000	300	54,000	72.0	165	180	100,800	100,800	100,000	Francis
Hiwassee	--	--	6,280	438,000	365,000	224	130,000	22.0	190	200	57,600	115,200	115,000	Francis
Cherokee[1]	--	--	32,200	1,640,000	1,473,000	360	300,000	58.5	100	110	90,000	120,000	120,000	Francis
Blue Ridge[3]	--	--	3,290	197,500	183,000	110	58,000	10.0	147	--	20,000	20,000	20,000	Francis
Ocoee No. 1[3]	--	--	1,380	76,700	25,900	362	--	7.5	110	--	18,000	18,000	18,000	Francis
Ocoee No. 2[3]	--	--	--	--	--	--	--	--	250	--	18,800	28,200	18,000	Francis
Great Falls[3]	--	--	2,290	55,100	49,900	450	150,000	--	142	--	29,400	29,400	29,000	Francis

1 Under construction.
2 Two lock chambers.
3 Acquired by purchase of completed projects; dependable flood storage not fully determined.
4 Generating capacities are based upon actual performance which exceeds guaranteed performance.

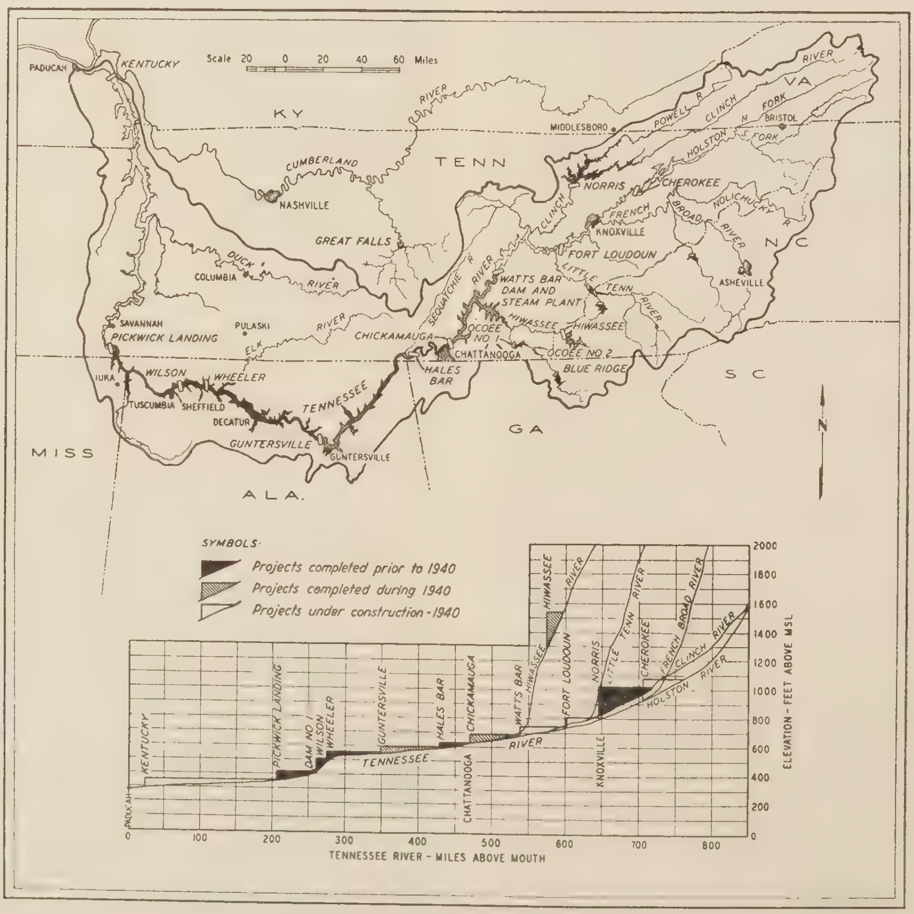


FIG. 1 MAP OF THE TENNESSEE VALLEY

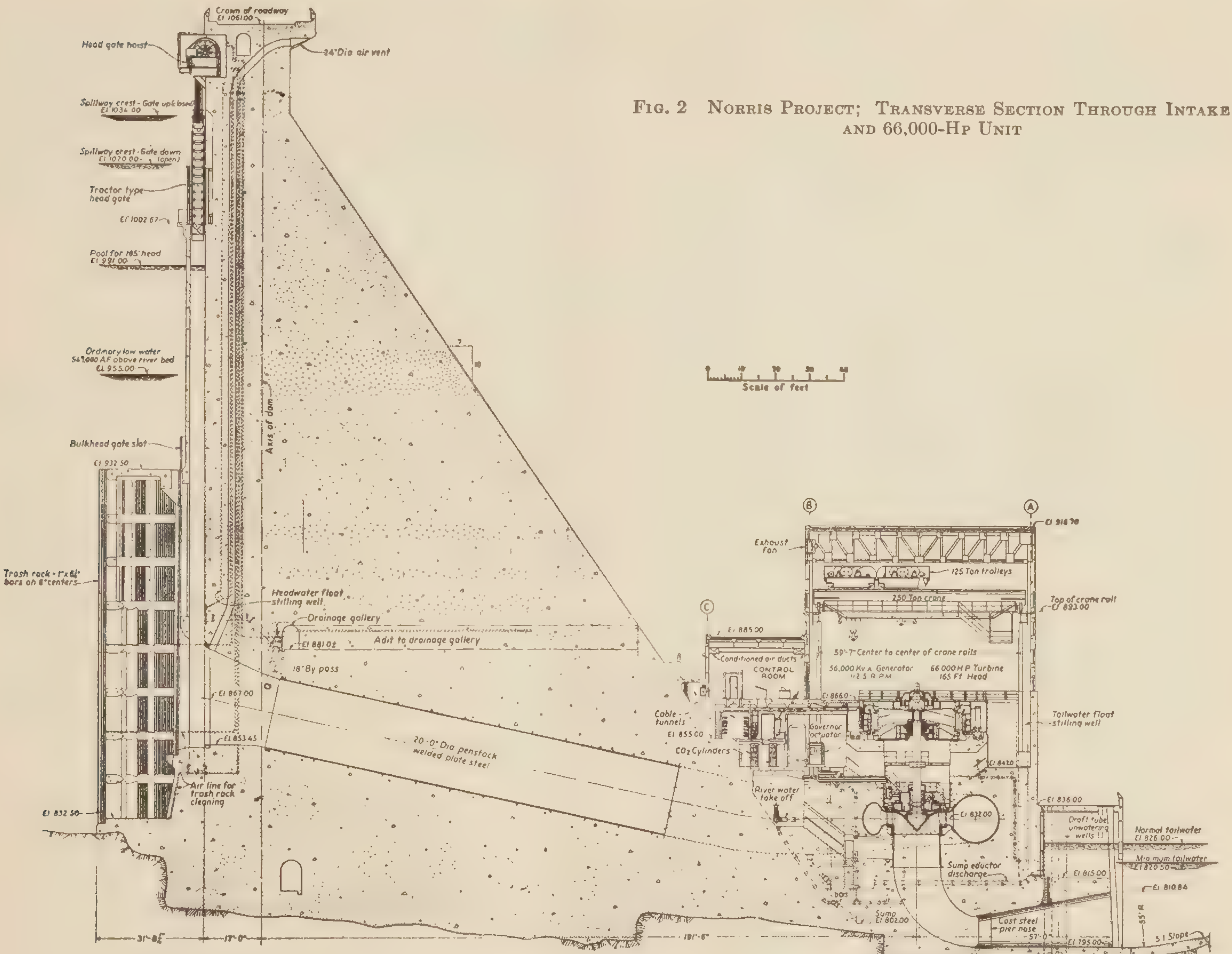


FIG. 2 NORRIS PROJECT; TRANSVERSE SECTION THROUGH INTAKE AND 66,000-HP UNIT

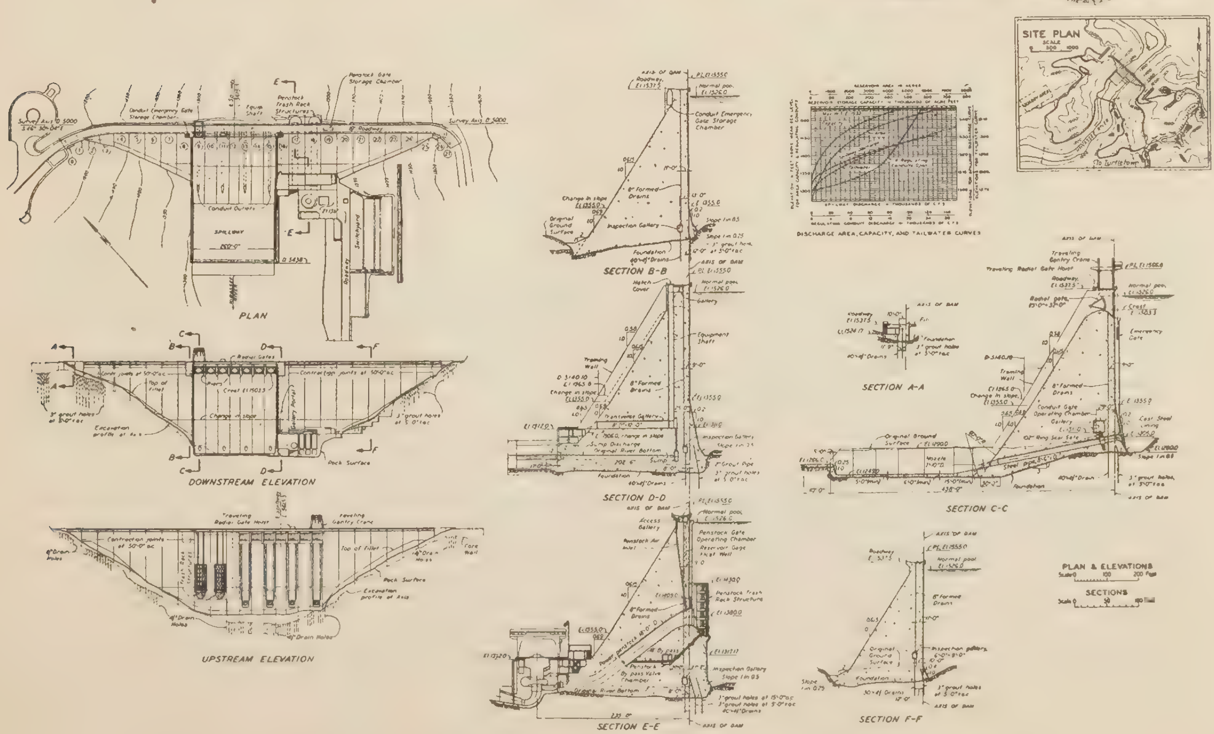


FIG. 3 HIWASSEE PROJECT; GENERAL ARRANGEMENT, PLAN, ELEVATION, AND SECTIONS OF DAM

TABLE 2 PHYSICAL FEATURES

	Norris	Hiwassee
Project started	Oct. 1933	July 1936
Machinery ordered	Oct. 1934	Sept. 1938
Machinery operating	July 1936	May 1940
Drainage area - sq mi	2912	966
Average run-off - cfs	4600	2100
Maximum recorded flood - cfs	115,000	42,000
Minimum recorded flow - cfs	200	134
Flood capacity - cfs - sluices	36,000	20,000
- overflow	54,000	130,000
- total	90,000	150,000
Number of sluices	8	4
Spillway gates	3-100'x14'	7-23'x32'
Elevation of gate sills	1020	1503.5
Pond levels - elevation and storage volumes - ac.-ft		
Ordinary minimum	955.....554,000	1415.....73,000
Normal storage	1020...2,047,000	1526.5..438,000
Controlled flood storage	1034...2,567,000	1532....474,000
Tailwater - normal elevation - ft	826	1272
- minimum elevation - ft	820.5	1266
Head - gross - maximum ft	194 (Elev.1020-826)	255 (Elev.1525-1272)
- minimum ft	129 (Elev. 955-826)	143 (Elev. 1415-1272)
Head - net for turbine rating - ft	165	190
Turbine horsepower and number of units	132,000 in 2 units	80,000 in 1 unit
Turbine discharge - rated capacity - cfs	8500	4500
Provision for additional capacity	None	A 2d 80,000-hp unit
Increase in dry river flow downstream - approximate cfs	6000	3200
Generator rating - kva	2 @ 56,000	1 @ 64,000
- kw	2 @ 50,400	1 @ 57,600

navigation conditions during periods of low flow in the lower Mississippi River.

The approximate flow required for interim 6-ft navigation between Pickwick and Paducah under present conditions is about 17,000 cfs. During this period of drawdown there is very little conflict between the requirements for navigation and power. The present electrical load of the Authority is largely industrial, relatively steady with a high load factor, and not subject to appreciable seasonal variations. Consequently, any water drawn out of storage to maintain a flow of 17,000 cfs in the lower Tennessee for navigation is utilized nearly 100 per cent for power generation. At the present stage of completion of the development as a whole, storage water released from both Norris and Hiwassee is used over a total head of 323 ft in the six main river plants already completed, not including the variable head at the storage dams.

On December 15, the operating schedule requires that the reservoirs be practically empty to accommodate flood storage between that date and April 15. This sometimes requires water to be drawn out of storage which would be held if operation were controlled solely by power requirements.

During the period from December 15 to April 15, the elevation of the reservoirs is limited to certain maximum levels, excess water being either wasted or used through the generating units at Norris and Hiwassee if the rate of discharge permits.

Determination of Turbine Capacity

When, in 1933 and 1934, studies were being made for the installation of generating machinery at Norris Dam, the size, type, and characteristics of the Authority's electrical system were somewhat nebulous. Wilson Dam and generating plant, with 182,000 kw installed, was the only operating station. Wheeler Dam was under construction, and one unit was contemplated as the initial installation. As will be seen from Fig. 4 the total load on the TVA system in 1934 was less than 50,000 kw, part of which was on short-time contracts to neighboring utilities and which could be taken or not at their option.

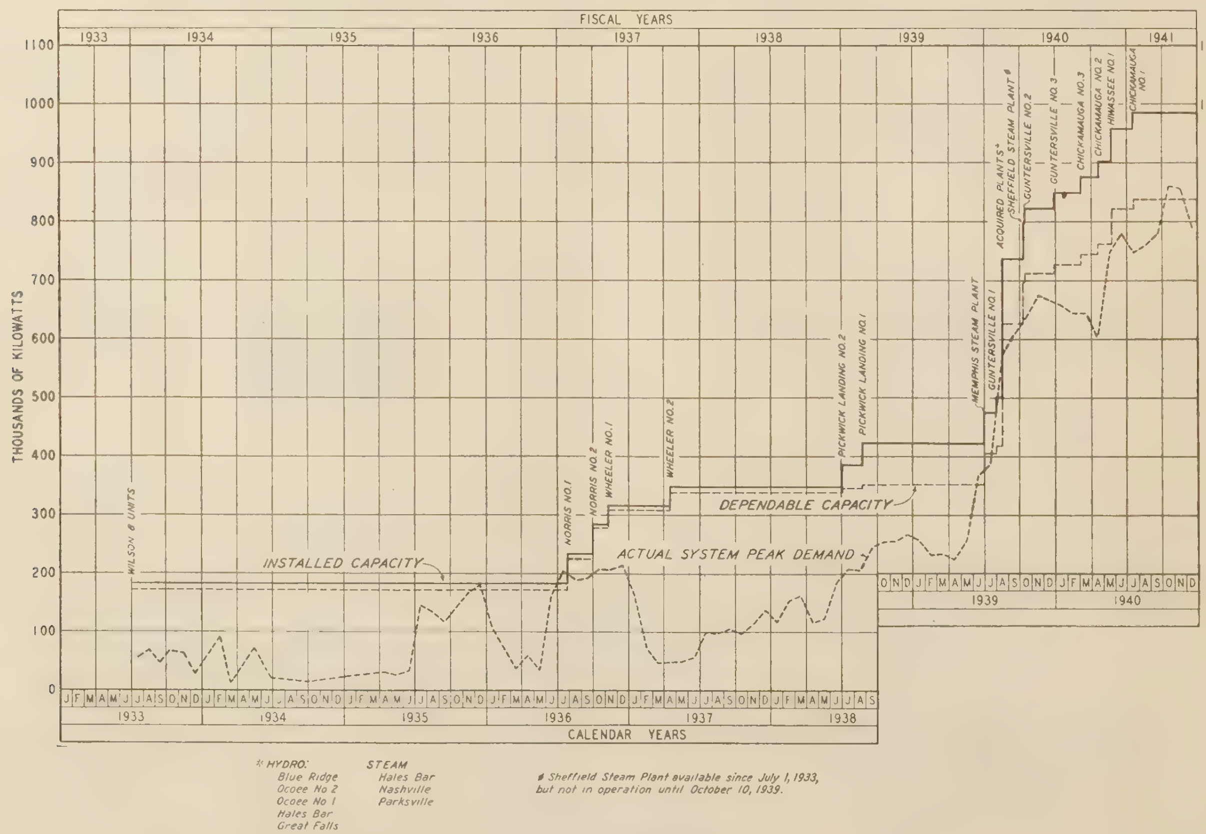


Fig. 4 Available System Capacity and Actual Peak Loads, 1933–1940

Based on the storage available, the maximum discharge necessary to improve the extreme low flows of the Tennessee, and future probable six-plant TVA system, with a total installation of approximately 525,000 kw, the capacity of the Norris plant was planned to be 100,800 kw, in two units rated 50,400 kw, 66,000 hp, 165-ft head.

In 1936, when funds were first appropriated for the start of construction on Fowler Bend Dam, later called Hiwassee, no power installation was planned initially, although provision was to be made for a future installation of two 40,000-kw units. The

TABLE 3 HYDRAULIC-TURBINE DATA

	Norris	Hiwassee
Rpm	112-1/2	120
Name plate rating - hp	66,000	80,000
Rated head - ft	165	190
Specific speed	48.8	48.1
Head for best efficiency - ft	180	200
Maximum usable hp	75,000	93,000
Maximum gross head - ft	194	254
Minimum gross head - ft	129	143
Elevation - center line of turbine - ft	832	1277
Elevation - normal tailwater - ft	826	1272
Runner inlet - inches	161	161
Discharge diameter - inches	165-1/2	165-1/2
Height - inches	41-7/8	41-7/8
Penstock diameter - ft	20	18
Penstock length - ft	154	220
Turbine scroll case inlet diameter	17'-8"	17'-3"
Scroll case design pressure - lb/sq in	125	140
Maximum scroll case plate thickness - inches	1-11/16	1-11/16
Governor cylinders - diameter - inches	28	28
- stroke - inches	13-3/4	13-3/4
Runaway speed at maximum head - rpm	215	235
Main shaft diameter - inches	35	36
Length of main shaft	14'-0"	12'-4"
Turbine bearing diameter - inches	36	37
Length of turbine bearing - inches	36	36
Weight of turbine runner and shaft - lb	153,500	142,500
Hydraulic thrust - lb	385,000	410,000
Weight generator rotor - lb	483,000	489,000
Total weight revolving parts - lb	1,104,000	1,106,000
Thrust bearing diameter - inches	74	76
Thrust bearing manufacturer	Kingsbury	Kingsbury
Powerhouse crane capacity - tons	250	275
Generator rating - kva	56,000	64,000
- kw	50,400	57,600
Generator WR^2 - lb ft sq	70,000,000	60,000,000
Reverse power required for motoring - kw	1350	1580

1936 report to Congress, made at about that time, indicates that four plants with an installed capacity of 348,000 kw and a continuous capacity of 225,000 kw were to be the nucleus of the TVA generating system and that at some possible future date machinery having a total continuous capacity of 660,000 kw might be installed in eleven plants. Fig. 4 shows how soon this expectation was surpassed.

Two 36,000-kw generators were authorized for Pickwick in the summer of 1936, and, in the summer of 1937, two 25,000-kw units for Guntersville and two 27,000-kw units for Chickamauga. In 1938, when the first Pickwick unit was nearing completion, overtime work was authorized to hasten the completion date so as to accommodate the increase in load. With continued steady growth of the TVA load, a third unit was authorized for both the Guntersville and Chickamauga plants. At that time, final studies were being made to determine the exact capacity of the generating units for Hiwassee Dam. The final decision was for an ultimate installation of two 57,600-kw units driven by 80,000-hp, 190-ft head, 120-rpm hydraulic turbines, although the use of three 40,000-kw units with an initial installation of two units was seriously considered.

The 115,200-kw ultimate installation increases that contemplated in preliminary studies by 44 per cent. This was due to the changed perspective resulting from the unpredictably rapid growth of the TVA system and the desirability of having larger units and greater installed capacity for peak loads and for spinning reserve in case of emergencies.

Features of Design

A fortunate set of circumstances made it possible to have the Hiwassee turbine almost an exact duplicate of the Norris turbines, both turbines being furnished by the Newport News Shipbuilding and Drydock Company. Only slight modifications were made, such as a larger shaft diameter and higher tensile strength in the plates of the spiral casing, together with slightly thicker metal in some other parts, in order to take care of the larger power output and higher head conditions. Table 3 shows the principal physical features of both the Norris and Hiwassee installations.

Much thought and discussion were given to the design of the

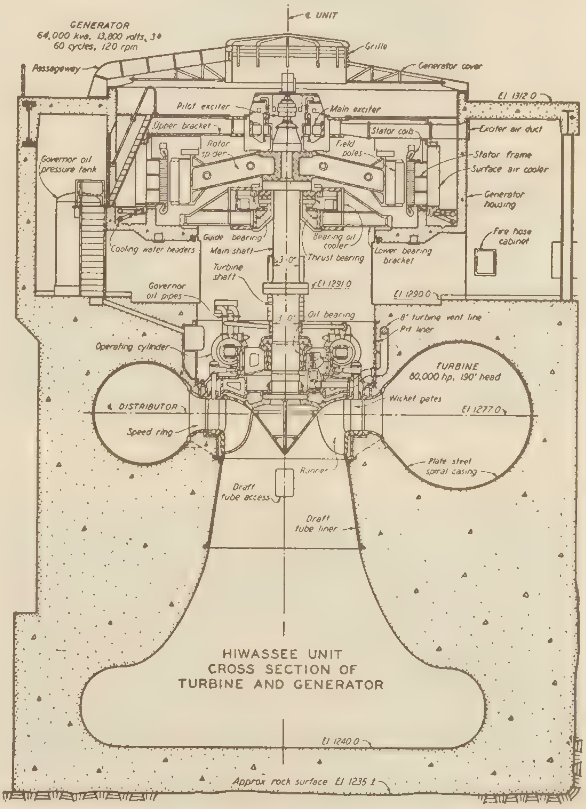


Fig. 5 Hiwassee Unit; Cross Section of Turbine and Generator

plate-steel scroll case, particularly at the terminal point of the volute, where the smaller areas usually require elliptical sections. The speed ring must be designed for the largest or inlet section of the scroll case. With a guide-vane height of $41^7/_8$ in. and a scroll-case inlet diameter of 17 ft 8 in. the speed-ring flanges have an angle of 50 deg with the horizontal and an outside width of 7 ft $1^3/_4$ in. Since the scroll-case area is decreased around the turbine as the water is fed into the runner, a point is reached about $^5/_6$ of the way around the circumference, where a circle of 7 ft diam would have the required area of $^1/_6$ of the inlet area. Obviously, the vertical height of the casing must be greater than the height of the speed ring in order that the 50-deg flanges of the speed ring (100 deg total included angle) shall come tangent to the circle. One method of reducing the area and still maintaining the vertical height of the casing is to make the scroll-case sections elliptical at the small end.

Elliptical sections, however, when subjected to internal pres-

sure, tend to become round and. in a complicated structure of this type, exact determination of the resulting stresses is difficult. In order to reduce these stresses the Newport News Shipbuilding and Drydock Company proposed using a special steel casting which forms the terminal section of the scroll case and also the junction between the terminal and inlet sections. While this has a complicated shape, it could be heavily ribbed and stiffened so as to form a rigid connection. This special casting was used on both the Norris and Hiwassee turbines. In addition, in view

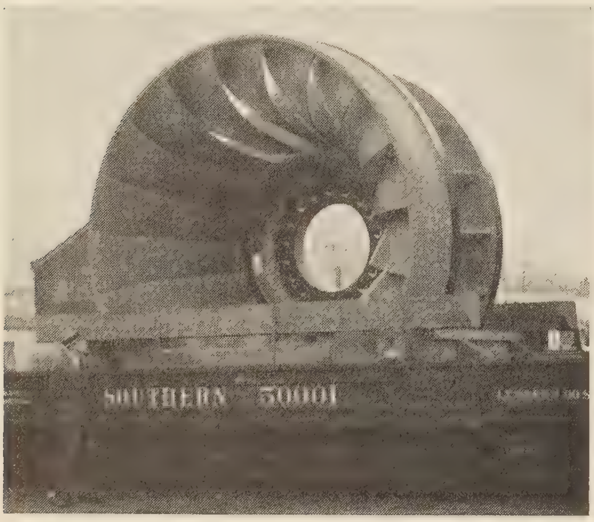


FIG. 6 161-IN. RUNNER FOR NORRIS DAM

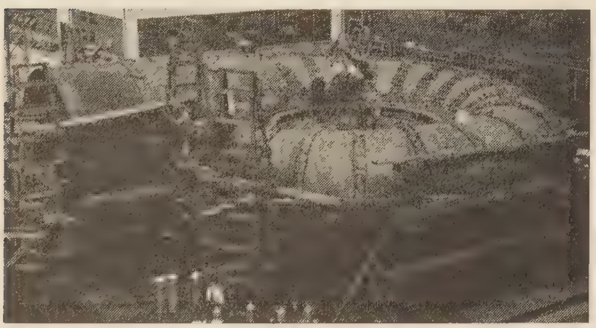


FIG. 7 SPIRAL CASING FOR 80,000-HP TURBINE FOR HIWASSEE DAM

of the impracticability of trying to hold the full stress in the casing metal in the small one quarter of the casing, that section was embedded solidly in concrete and sufficient reinforcing added to assist the plate-steel structure.

Over the balance of the scroll case, from the inlet around approximately three quarters of the circumference, a layer of cork and felt $^3/_4$ in. thick is laid over the plates so that the expansion of the casing under pressure will not be transmitted to the concrete.

The Norris scroll case, with an inlet diameter of 17 ft 8 in. and designed for a pressure of 125 psi, used a maximum plate thickness of $1^{11}/_{16}$ in. with $1^1/_4$-in. rivets. Even rivets of this size are difficult to drive by hand, so when the Hiwassee scroll case, of practically the same size and for a pressure of 140 psi, was under discussion, it was decided to use a higher-strength steel so as not to require thicker plates or rivets larger than $1^1/_4$ in. diam.

While the harder plate and harder rivets increased the difficulties of assembly and riveting somewhat, a tight job was obtained. An interesting method of testing joints was developed and consisted of smearing soapy water on the outside of a joint and directing a stream of compressed air at 100 lb pressure at the inside. Any bubbles indicated leaks which required additional calking.

On the Hiwassee turbine, in order to guard against corrosion and rusting of the guide-vane stems, where they pass through the grease- and water-lubricated bearings adjacent to the waterway, a special hard bronze bushing or sleeve $^1/_4$ in. thick was shrunk on the guide-vane stem in addition to the usual bronze bushing in the head cover and curb plate.

Similarly, the top and bottom of the guide vanes were protected by a piece of 12 to 14 per cent chromium-steel plate $^1/_2$ in. thick cut to the exact profile of the guide vane and welded to both the top and bottom of the guide vanes. The facing plates adjacent to the guide vanes on both the head cover and the curb ring were also of 12 to 14 per cent chromium-steel plate, $^3/_4$ in. thick. The clearance between the ends of the guide vanes and the stationary plates was about 0.015 in. top and bottom. It was very gratifying, after 7 months' operation of the Hiwassee turbine, to note the bright clean surfaces of these chromium-

TABLE 4 COMPARISON OF SCROLL-CASE FEATURES

	Norris	Hiwassee
Casing design pressure - lb/sq in	125	140
Steel used - ASTM	A89-33, Grade B Flange, low tensile carbon steel	A149-36, Grade A Firebox quality silicon steel
Ultimate strength - lb/sq in	50,000	65-77,000
ASTM designation - Rivets	Boiler rivet steel, A31-24	Structural rivet steel, A141-39
Ultimate strength - lb/sq in	45-55,000	52-62,000
Allowable stress in spiral casing - lb	12,000	14,000

steel facing plates as contrasted to the usual rust-pitted surface where the ordinary plate steel is used.

Both the stationary and rotating clearances adjacent to the runner are protected with steel wearing rings, the stationary rings being of hard silicon steel, those on the runner being of mild steel.

When the Norris turbines were purchased, it was felt that a more accurate alignment of the runner clearances could be obtained by grinding the stationary wearing rings in the head cover and curb ring after the spiral casing and speed ring were embedded in the concrete. For this purpose a special grinding ring was supplied with the turbines, and, after the concrete had set and most of the heat had been dissipated, these clearances were very accurately ground out.

The design clearance between the runner crown and the head cover is 0.040 in. radially and 0.060 in. at the runner band. At Norris this grinding was completed about February 1, 1936, before any water had been impounded in Norris Lake. Filling of the reservoir started early in March. Early in June it was found that these runner clearances had shifted materially, the tight side being upstream. With no contraction joint between the power station and dam, the water load against the dam caused a slight tilting of the powerhouse and a deformation of the substructure, which forced the stationary parts of the turbine slightly out of round. While it would have been possible to remove the internal parts of the turbine and regrind the clearances, this was not considered necessary. A slight realignment of the bearings was made and the unit put into operation, with satisfactory results. The minimum runner clearances after readjusting the bearings was about 0.025 inch.

In view of the building settlement and resulting distortion of

the stationary rings at Norris, the grinding device was not used at Hiwassee, but the parts were all machined to final dimensions in the shop. However, the proper contraction joint between the dam and powerhouse was provided at Hiwassee, and no appreciable distortion occurred in the Hiwassee turbine parts due to deflection from the water load on the dam.

The Norris penstocks are 20 ft in diam and have field-welded circumferential joints. The Hiwassee penstocks are only 18 ft in diam and have riveted joints throughout. The smaller size of the Hiwassee penstocks is justified because of the lower capacity factor at which the Hiwassee plant will operate after the second unit has been installed. Furthermore, as additional storage is provided, the period over which that storage must be utilized becomes longer, resulting in yet lower capacity factor even during the period of drawdown. The total entrance loss, penstock friction loss, and rack loss at Norris for a flow of 4472

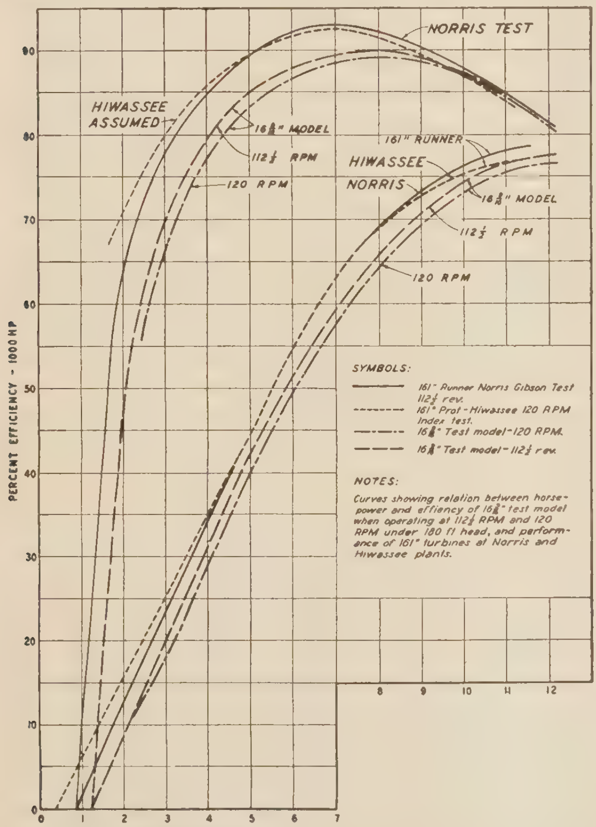


Fig. 8 Curves Showing Horsepower and Efficiency of Norris and Hiwassee Turbine Plotted Against Gate Opening

cfs was 0.6-ft head. At Hiwassee, the corresponding loss amounted to 1.95-ft head for a flow of 4470 cfs. This difference of 1.35 ft amounts to 0.71 of 1 per cent at full load but reduces to 0.56 ft or 0.3 of 1 per cent head at best efficiency load.

The stress-relieving system used for the field-welded circumferential joints in the Norris penstocks consisted of electric heaters, the heat being applied to the area about 10 in. on either side of the weld. While this procedure removed all stresses from the neighborhood of the weld, it apparently set up rather severe strains at the edge of the heated section. As a result, field stress relieving was discontinued, but a thorough peening of the weld metal was employed, after laying in each bead, and a satisfactory job was obtained.

Efficiency Tests on Norris Turbines

Efficiency tests on the Norris turbines were conducted after installation was completed, using the Gibson method of water measurement, which is based upon the equation of impulse and momentum. The differential system was used wherein the instantaneous difference between changes in pressure between two fixed cross sections in the conduit was recorded. The cross sections with four piezometer taps at each section were 91 ft 10 in. apart in the straight part of the 20-ft-diam penstocks.

TABLE 5 TURBINE-EFFICIENCY TESTS: NORRIS PROJECT

	Guaranteed Performance	Test Results Unit 1	Unit 2
Performance at 60,000 hp efficiency - percent	91.0	93.1	93.3
Maximum efficiency - percent	91.0	93.2	93.3
Hp at maximum efficiency	60,000	62,000	60,000
Maximum output - hp	75,200	78,300	78,300
Efficiency at 75,200 hp - percent	86.0	88.9	89.9
Efficiency at maximum output - percent		84.1	84.0
Discharge - cfs, maximum	4275	4560	4562
Generator efficiency - percent			
100 percent p.f. - 56,000 kva	97.75	98.25	98.25
90 percent p.f. - 56,000 kva	97.3	97.97	97.97

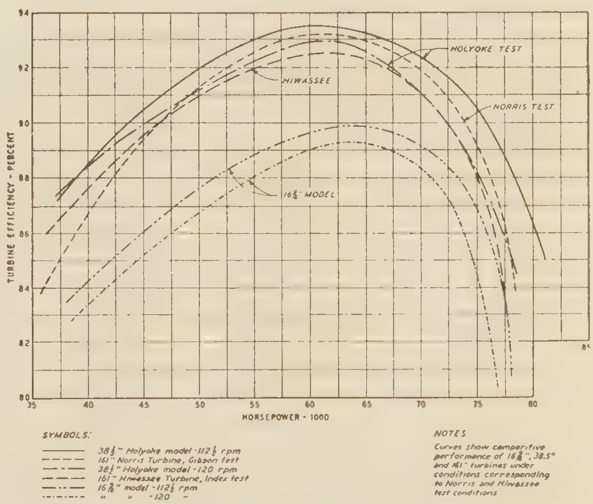


Fig. 9 Efficiency Curves of Norris and Hiwassee Turbines Plotted Against Horsepower

The head measurement was obtained from float-operated headwater and tailwater gages located in stilling wells near the intake gates and the tailrace. The corrections for rack loss and penstock friction were obtained by differential manometers connected to piezometers located just downstream of the trashracks and at the entrance to the turbine scroll case.

The power output of the generator was measured by two sets of calibrated wattmeters connected to the generator terminals. Since the generators had been tested for efficiency, the turbine output could be obtained by dividing the generator output by the known generator efficiency. These tests were conducted in accordance with the A.S.M.E. 1926 Power Test Code.

The results of the efficiency tests on the Norris turbines are shown in Table 5 and in Figs. 8 and 9, the performance of the two units being so nearly alike that the difference is not perceptible. These data are for 180 ft net head.

Fig. 8 shows a comparison between the performance of a $16^9/_{16}$-in-diam model runner tested at the laboratory of the Newport News Shipbuilding and Drydock Company and the 161-in-diam Norris turbine. Horsepower at 180 ft head and efficiency are plotted against inches of opening of the wicket gates. While the horsepower output at any given gate opening shows an increase

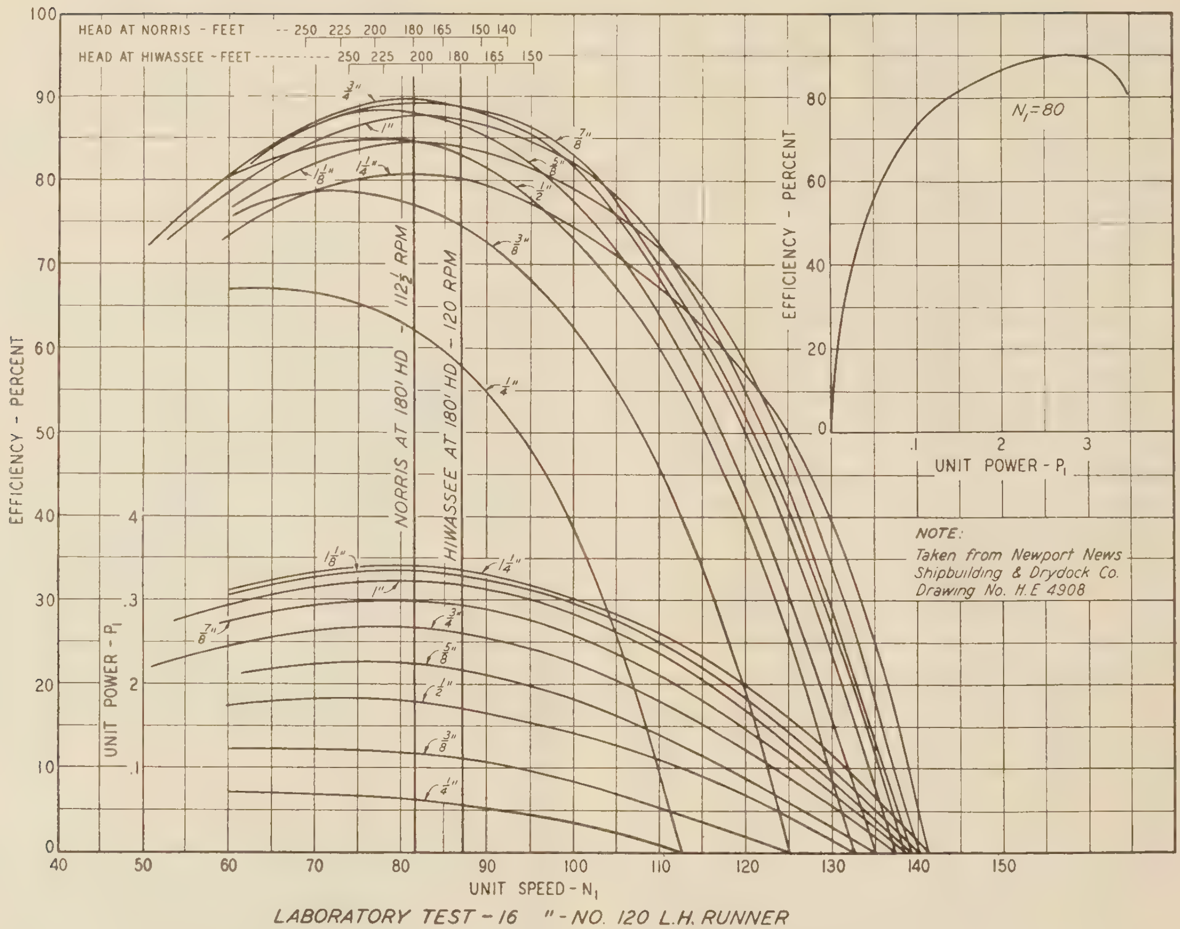


Fig. 10 Test of $16^9/_{16}$-In-Diam Runner at the Laboratory of the Newport News Shipbuilding and Dry Dock Company

of approximately 5 per cent when comparing the 161-in. prototype at Norris with the $16^9/_{16}$-in. test model, the maximum output only increased about $1^1/_2$ per cent, although the 161-in. prototype required less opening of the wicket gates to produce the same output. The laboratory test on the $16^9/_{16}$-in. model runner is shown in Fig. 10.

The 161-in. Norris runner also shows the greatest gain in efficiency at the smaller power outputs. This may be due in part to the effect of the spiral casing, as the $16^9/_{16}$-in. test model was not tested with a spiral casing.

Fig. 9 shows the comparison between the performance of the $16^9/_{16}$-in. model and the 161-in. Norris runner plotted with efficiency against horsepower. On this basis, the increase in efficiency at part load, while appreciable, is less noticeable than when plotted on the other basis. Plotting efficiency against horsepower is the most generally accepted method of comparison. Large turbines in scroll-case settings usually develop their power at a smaller wicket-gate opening, owing probably to the effect of the scroll-case velocity at the entrance to the wicket gates.

Fig. 9 also shows the performance of a 38.5-in. test model at the Holyoke testing flume. This test was made in 1926, on a homologous runner, but with a straight-tapered draft tube and with a different number and type of wicket gates, so that the performances are not exactly comparable. The Holyoke test shows a maximum of 93.6 per cent efficiency, as compared with 93.3 per cent at Norris.

To date no actual efficiency tests have been made on the Hiwassee turbine, although index tests show very closely the performance of that unit. As at Norris, the tests were made with a gross head of between 175 and 185 ft, but since the Hiwassee unit operates at 120 rpm as against a speed of $112^1/_2$ rpm for the Norris turbines, the results cannot be compared directly, but a reasonable comparison may be obtained by taking the performance of the $16^9/_{16}$-in. model at the different unit speeds, which are approximately 6 per cent apart.

Fig. 8 shows the difference in the horsepower and efficiency of the $16^9/_{16}$-in. model turbine plotted against inches of gate opening at the two different speeds. In all cases, the Hiwassee conditions require greater gate opening for the same horsepower and give about 1.5 per cent less output at full gate. The efficiency curve for the Hiwassee conditions is also about 1 per cent lower for all gate openings. It should be pointed out that, while the Hiwassee turbine is operating at a lower efficiency than the Norris turbine under the particular test condition of 180 ft head, this is not true throughout the full range of operating heads. The Norris turbines were designed for best efficiency under about 185-ft head, and the tests were run at about 97.5 per cent of this head. The Hiwassee turbine operates at best speed under about 212-ft head, and the tests were run at only 85 per cent of this head, hence, the difference in efficiency for the test conditions.

The 161-in. Hiwassee turbine shows the same output for gate openings between $4^1/_2$ and 7 inches. At full gate it shows about 2 per cent less power, and at small gate openings it shows up to 5 per cent greater output than the Norris turbine, as shown in Fig.

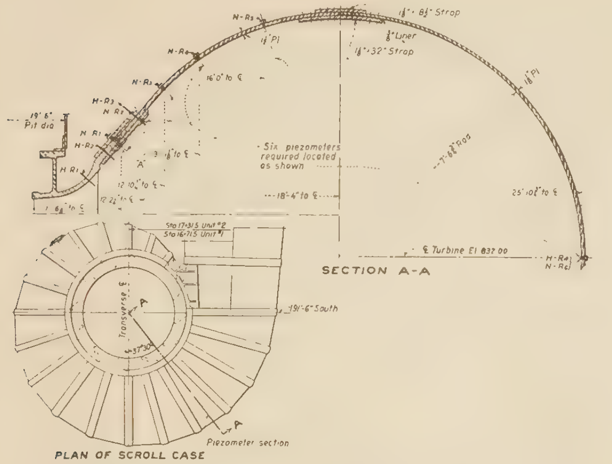


FIG. 11 NORRIS POWER PLANT; PIEZOMETER LOCATION

8. The increased output at the small gate openings is probably accounted for by the fact that during the Hiwassee tests the air valve, admitting air to the space between the runner and head cover and then through the runner into the draft tube, was blocked wide open at all gates. Originally this was operated by a cam which started to open the valve at approximately 40 per cent gate and held it wide open at 30 per cent gate and below. Some preliminary tests indicated that quieter operation and greater output could be obtained with the turbine vented even at considerably larger gate openings.

A study of the horsepower-gate curves of the $16^9/_{16}$-in. model and of the Norris and Hiwassee tests indicates that, under the test conditions of 180 ft head, the efficiency of the Hiwassee turbine should be slightly less than the efficiency of the Norris turbines.

All the turbines installed by the Authority are provided with the Winter-Kennedy type of differential pressure piezometers in the scroll case. Fig. 11 shows the arrangement of these piezometers for both the Norris and Hiwassee turbines. During the Gibson tests on the Norris turbines readings were taken on water columns connected to each of these piezometers; and after the discharge had been accurately determined, it was possible to obtain an equation for any set of piezometers reading as follows:

cfs = KD^N where
cfs = discharge of turbine in cubic feet per second
K = constant obtained experimentally
D = head difference in feet of water between piezometer on outside of scroll case, called R_6 in Fig. 11, and one of the piezometers on inside of scroll case, R_1, R_2, R_3, or R_4, Fig. 11.
N = coefficient, usually very close to 0.5, as in venturi-meter formula

The equations for two sets of the piezometers at Norris, based on the water measurements by the Gibson method, were as follows:

For unit 1 .... $\begin{cases} \text{cfs} = 1334\ D^{0.509} \text{ for } R_6\text{-}R_1 \\ \text{cfs} = 1652\ D^{0.503} \text{ for } R_6\text{-}R_3 \end{cases}$

For unit 2 .... $\begin{cases} \text{cfs} = 1293\ D^{0.509} \text{ for } R_6\text{-}R_1 \\ \text{cfs} = 1620\ D^{0.510} \text{ for } R_6\text{-}R_3 \end{cases}$

It will be noted that the coefficient N is very close to the 0.5

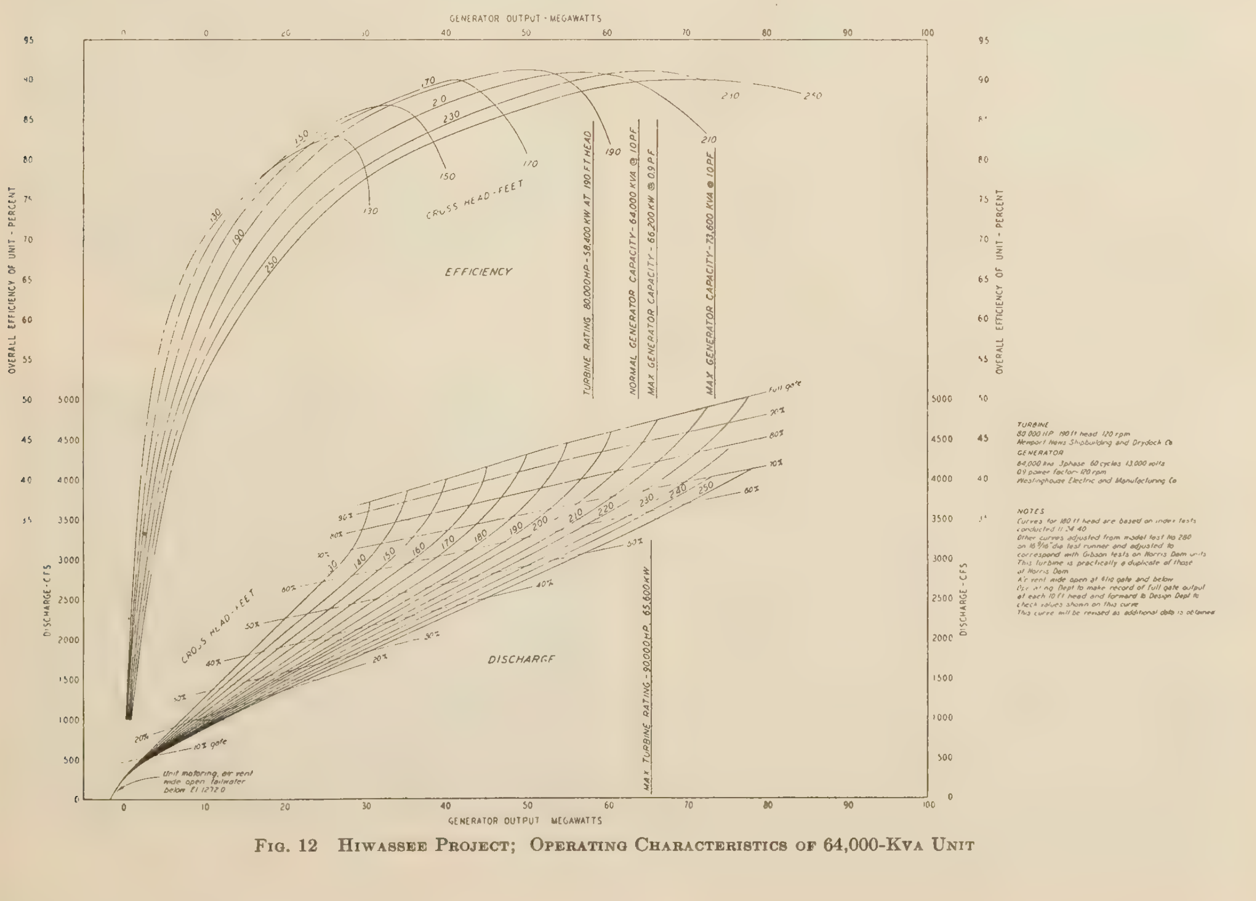


FIG. 12 HIWASSEE PROJECT; OPERATING CHARACTERISTICS OF 64,000-KVA UNIT

power; and, in accordance with the conclusion of J. M. Mousson,[3] the square root may be used without appreciable error.

Where no accurate water measurements are made, it is necessary to arrive at the coefficient K in a different manner, the exponent N being assumed to be 0.5. The generally accepted method is to plot a curve of hp/$D^{1/2}$, this curve having the general shape of the over-all-efficiency curve of the unit. After determination from an inspection of the horsepower-gate curves in Fig. 8, that the maximum turbine efficiency of the Hiwassee turbine under 180 ft head should be about 1 per cent less than at Norris, or about 92.3 per cent, the discharge for the maximum efficiency can be computed and from this the coefficient can be derived for the discharge formula. In the case of the Hiwassee turbine, this was computed to be $Q = 1535D^{1/2}$. Calculating the discharge and efficiency for the Hiwassee turbine, we obtain the curve shown in Figs. 8 and 9, which lies slightly above the Norris efficiency curve for part loads but lies below the Norris curve at maximum efficiency and at full load.

The Department of Operations of the Authority uses a curve, such as that shown in Fig. 12, giving over-all efficiency and discharge against kilowatt output for various heads between 130 ft minimum and 250 ft maximum. The curve for 180 ft head is taken directly from the index test, but the other curves are based on the $16^{9}/_{16}$-in. model test adjusted in accordance with the field performance; in this particular case, upon the performance of the Norris turbines at various heads.

[3] "Water Gaging for Low-Head Units of High Capacity," by J. M. Mousson, Trans. A.S.M.E., vol. 57, 1935, pp. 303–316.

Discussion

R. V. Terry.[4] It is seldom that a manufacturer has the opportunity of furnishing large turbines of identical design for two or more power plants. In this case, the three Norris and Hiwassee turbines are practically identical, as mentioned by the authors. A fourth turbine of the same design was furnished for the Flathead plant of the Rocky Mountain Power Company in Montana, rated 77,000 hp, 189 ft head, 112.5 rpm.

Fabrication of the heavy steel-plate casings was a major job, especially the casing for Hiwassee which was made from the higher-tensile-strength material. The casings were built by men originally trained in the construction of Scotch boilers for marine use. A brief outline of the procedure for building a casing of this type is as follows: Each plate, which is of irregular shape, is developed one quarter size in the drawing room. Full-size molds for laying off the plates are then made from offsets, following a procedure generally used in shipbuilding. After laying out, the plates are cut to size and partially drilled ($^{1}/_{16}$ in. undersize) for the fitting-up bolts. The plates are rolled to the correct curvature and the butt straps are knuckled where necessary on a hydraulic press. Adjoining plates are fitted together and the joints ironed out locally as necessary on a hydraulic press. After each plate has been temporarily fitted to its adjacent plate, the complete shop assembly with the speed ring is started, beginning with the large end of the casing. Considerable use is made of heavy jacks, shoring, chain falls, bolting machines, etc.

After all sections of the casing are in place around the speed ring, the joints are laid up by local heating as necessary, and by a liberal use of fitting-up bolts. The rivet holes are then drilled to size. The general alignment of the casing is checked several times during the assembly processes, but the final center lines are not established until the shop assembly has been completed. After the assembly is dismantled, the speed ring is finally machined and drilled for its several other connecting members, using the center lines established from the assembly. In this way, a correct field assembly is assured. Each of the drilled rivet holes serves as a dowel hole so that no difficulty is experienced in relocating the plates in the field assembly exactly as they were in the shop. No rivets are driven until all parts of the casing are assembled and aligned.

The special cast-steel piece at the junction of the small and large ends of the casing resulted in some simplification in design and economy in construction. This part was actually made in two pieces, one at the top and one at the bottom, joined by steel plates.

Some of the $1^{1}/_{4}$-in. rivets at the speed-ring flanges passed through four thicknesses of material and had a grip length of 5 in. The rivets were driven with 90-lb hand guns, using dies with a special shaped shank to prevent breakage. Standard rivet sets with straight shanks were found to be unsuitable for this service. It was also found from experience in driving the rivets that the longer rivets had to be made longer, by amounts up to $^{5}/_{8}$ in., than the lengths determined by conventional methods.

The main-shaft stuffing boxes, which were about 40 in. diam, were packed with six rings of $1^{1}/_{4}$-in-square braided asbestos packing, lubricated with graphite. A lantern ring for the admission of cooling water was placed at the middle of the box. No grease is used in the boxes as previous experience with similar large units demonstrated that better service was secured by omitting all lubricants other than the graphite and water. When grease was used, the packing rotated and tended to jam and cause excessive heating.

The main-bearing oil sump is located in the crown plate where it is cooled by the adjacent water. No cooling coils were found necessary. The maximum temperatures were found to be about 130 F.

Some of the more important weights of each turbine were approximately as follows:

Runner	45 Tons
Turbine shaft	25 Tons
Speed ring	65 Tons
Crown plate	30 Tons
Spiral casing	225 Tons

The total weight of one complete turbine, exclusive of governor equipment, was about 450 tons.

The Holyoke model tests were made in 1926. The 93.6 per cent efficiency obtained on these tests is on a par with the best efficiencies ever obtained in that testing flume. However, while comparative, the Holyoke results must be discounted somewhat and cannot be stepped up in the customary way as is done with the more accurate results secured in modern hydraulic laboratories. It is often considered fortunate if the Holyoke efficiencies are equaled by the field results.

The laboratory model really should have included a model of the spiral casing. A riveted casing of the Norris and Hiwassee types probably results in relatively higher frictional losses near full load, which prevents full realization of the maximum power output that might otherwise be obtained. This is borne out by the comparative results shown in the authors' Fig. 8, where it is seen that very little gain in power or efficiency at the higher gate openings was obtained in the field, as compared with the model results. If the model had included a duplication of the casing construction, it is believed a greater differential would have been noted near full load.

[4] Hydraulic Engineer, Newport News Shipbuilding and Drydock Company, Newport News, Va. Mem. A.S.M.E.

The medium specific speed employed with these turbines results in a very satisfactory power-efficiency curve that is quite flat, and approaches the shape of curve that can be realized with the newer adjustable-blade type turbines suitable for the lower heads up to about 80 ft.

R. E. B. SHARP.[5] The Norris and Hiwassee casings were, presumably, completely riveted in the field, with the sections nested into each other for compactness during shipment. The external girth butt straps, it is noted, were located at the section joints and were, therefore, crimped to suit the angles between the sections.

As a contrast to this design are shown Figs. 13 and 14 of this discussion, a shop view of a welded-plate steel casing for one of the I. P. Morris turbines for the Lower Colorado River Authority, for installation in the Marshall Ford Plant near Austin, Texas. The turbines are rated at 27,000 hp under a head of 120 ft, with a maximum head of 216 ft. The casings were fabricated in the Baldwin Locomotive Works at Eddystone, Pa. The diameter at intake to the casing is 172 in., with a maximum plate thickness of $1^{5}/_{16}$ in. This thickness exists in the vicinity of the intake near the speed ring, while that at the intake around the periphery is only 1 in. The difference is due to considering the casing as a torus with the greatest radial loads near the center.

[5] Chief Engineer, I. P. Morris Department, Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

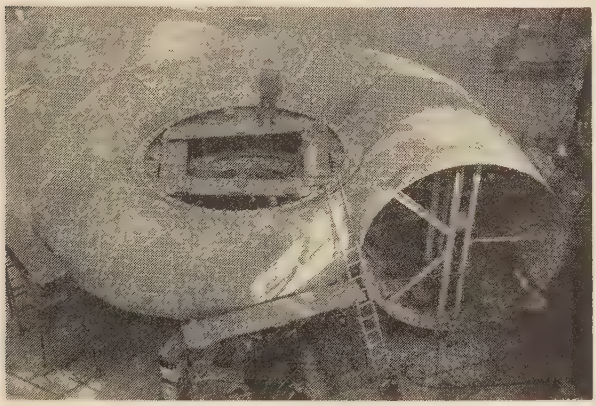


FIG. 13 WELDED-PLATE; TURBINE CASING

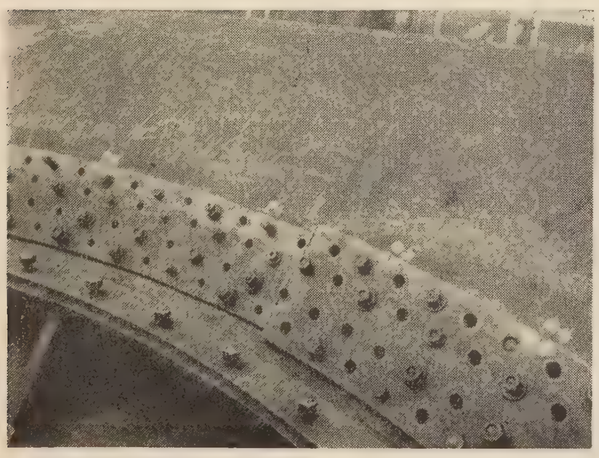


FIG. 14 DETAIL OF FIG. 13

As will be noted from Fig. 13, there are six joints for field riveting. These are located between sections and, therefore, no crimping was required of the butt straps.

All welds were radiographed, and were chipped out and rewelded where the films showed more defects than permitted by A.S.M.E. Class 1 weld. The sections were then stress-relieved, and finally fitted to the speed ring and to each other.

With a given allowable stress, this construction permits the use of thinner plate than does riveting, due to the superior efficiency of the butt-weld joint. The plate thickness at the speed ring, however, is determined by the riveted joints at that location. Also smoother flow surface is obtained than with girth-lap joints, which would have been employed otherwise; rivet heads are also avoided.

A different treatment of the terminal section of the casing from that described by the authors is illustrated in Fig. 13, which shows the section to be constructed entirely of plate. The outside wall of the casing is continued past the baffle vane to a welded junction with the intake portion, while a vertical baffle plate inside of the scroll curves inward to meet the cast baffle vane of the stay ring. The hollow space between the baffle plate and the outside wall is filled with grout, which effectively supports the baffle plate and retains its shape.

It is the writer's experience that had the Norris model been tested with a casing, the efficiency and power developed therefrom would have been slightly reduced. This would have had the effect of increasing above 1.5 per cent the relative increase of the prototype power above the model. It is also the writer's experience that the coefficient of flow through the wicket gates of the prototype is greater than for the model, even where a casing is employed in both instances. In making close comparisons of models and prototypes involving gate opening, it should also be borne in mind that the true value of ϕ tends to be lower on the prototype than on the model, due to the higher efficiency of the former.

A further explanation for the slight discrepancies between the model and prototype performances might lie in slight differences in design which result in the model and the large unit not being exactly homologous. It is noted that the power at best efficiency of the prototype is about 2 per cent less than that of the model. If the power-efficiency curve of the large unit is shifted to the right by this amount, the curve becomes very closely parallel with that of the model. Similar consideration would, to some extent at least, account for the convergence of the power-gate curves of the prototype and model.

It is believed that the authors' curve of the Norris test, Fig. 9, is incorrectly low at the smallest values of power, as in this respect it does not appear to check Fig. 8. The Holyoke test curves, while probably too high all along the line, do show the gain in efficiency at maximum and minimum values of power, due to the straight tube.

It is believed that the Hiwassee turbine is suffering a loss in power at the higher gate openings (authors' Fig. 8), due to air admission. The model curves, Fig. 11 of the paper, do not justify this relative decrease in power as compared to Norris. Air admission at full load generally results in some loss of power, but in smoother operation.

AUTHORS' CLOSURE

Mr. Terry calls attention to practical obstacles attending the use of $1^{1}/_{4}$-in. rivets with very long grip. The field erection forces experienced much difficulty in maintaining sufficient heat to insure workability in driving without actually burning the rivet steel.

Mr. Sharp gives an interesting description of the welded-plate-steel casing for the Lower Colorado River Authority. Such pro-

TABLE 6 POWER READINGS WITH AIR VENT CLOSED AND OPEN

Turbine gate opening	Gross head, ft	Air vent	Kw by watthour meter	Kw at 180 ft gross head	Loss power, air valve open	Power loss, per cent
100	181.66	Closed	55861	55200	...	...
90	181.58	Closed	54800	54100	.	...
80	181.63	Closed	52650	52000		.
70	181.74	Closed	49200	48500		. .
60	181.92	Closed	43650	43000	. .	
50	182.47	Closed	35950	35250	. .	. .
40	182.76	Closed	27330	26750	. .	...
60	182.08	Open	43650	42950	50	0.116
50	182.26	Open	35600	34950	300	0.86
40	182.90	Open	27250	26600	150	0.56
40	184.37	Open	27300	26350	400	1.52
50	183.61	Open	35780	34720	530	1.52
60	182.87	Open	43950	42950	50	0.116
70	182.45	Open	49400	48400	100	0.21
80	181.68	Open	52950	52250	—250	—0.48
90	181.56	Open	54600	53900	200	0.37
100	181.39	Open	55650	55000	200	0.36

cedure was unheard of in 1934, when the Norris turbines were ordered; and, even in 1938, when the Hiwassee turbine was awarded, there was no actual experience on welding the scroll case of a turbine of this size. A paper describing in detail the construction of a welded-plate scroll case and of the X-ray and pressure tests would undoubtedly be of great interest and value.

Mr. Sharp is correct regarding the lower part of the Norris test curve in Fig. 9 of the paper. This curve is in error from 50,000 hp down to 35,000 hp and should cross the 86 per cent line at 35,000 hp, which would bring it just above the Hiwassee curve.

Referring to the last paragraph of Mr. Sharp's comment, the authors submit Table 6 of this closure, giving power readings taken at Hiwassee with the air vent closed and open. Omitting the reading at 80 per cent gate, which is probably in error, the tests indicate that, with the air valve open at gates from 60 per cent to 100 per cent, there is a decrease in output of from 50 to 200 kw, or from 0.1 per cent to 0.37 per cent, which is too small to show on the curves of Figs. 8 or 9.

The Separation of Liquid From Vapor, Using Cyclones

By ARTHUR POLLAK[1] and L. T. WORK,[2] NEW YORK, N. Y.

In this paper, the various types of cyclones are reviewed and the data on their performance are summarized. Experiments with cyclones for separating liquids from vapors are described. The performance data obtained conform to the equation $\log [L_v (V/L)^a] = b + cV$; where L_v is the entrainment or liquid remaining in the effluent vapor, V is the vapor velocity, L is the liquid velocity, all in units of pounds per minute per square foot of inlet-duct cross section. Terms a, b, and c are constants associated with the equipment and its mode of operation. The principal source of the entrainment is shown to be the wall creep of liquid in the cyclone, impelled by the radial component of the vapor velocity.

Introduction

The separation of liquids from vapors is frequently a serious problem in industry. The performance and capacity of boilers, evaporators, gas scrubbers, and distillation units are often limited for a lack of adequate means for removing entrained liquid from vapor streams. Separators operated with such vapor-liquid streams frequently function at comparatively high vapor velocities, and often, large quantities of liquid must be separated. Cyclones are widely used for this work.

The performance of cyclones is governed predominantly by variables of two categories; (*a*) operating variables, relating to the properties and to the states and rates of the solids, liquids, and gases handled, and (*b*) design variables, involving the types, arrangements of parts, and dimensions of cyclones.

The number of operating variables is reduced considerably if interest be limited to the field of greatest practical importance; i.e., the separation of steam from aqueous solutions.

Design variables are affected in some measure by these operating variables. Thus, the velocity and relative amounts of liquid and of vapor influence the design of a cyclone. Very few systematic series of experiments have been published on cyclone design variables, and those only on dust separation. The cyclones most commonly used today are of simple construction, containing few parts. Simple cyclones are, therefore, of considerable practical interest and are the most likely to yield performance data capable of correlations which would be comparatively independent of physical dimensions.

If some relation among the separating efficiency of a specific cyclone, the velocity of the vapor, and the amount of liquid could be established and, perhaps, extended to other cyclones and vapor-liquid streams, an insight into the nature of the entrainment might be gained and some useful generalizations might be achieved.

While some experimental data have been published on the separating efficiency of cyclones on dusts, no data have been discovered on the effectiveness of cyclones for separating liquids from vapors.

Types of Cyclones

In reviewing the many variations in cyclone design, a system of classification appeared to be desirable. Accordingly, cyclones were classified in terms of the relative complexity of the path of an element of the gas or vapor stream. Such a grouping closely parallels the structural variations. With this functional definition in mind, the following classes were identified; examples of the various groups are also being considered:

1 "Simple Cyclones:" In this type the path of the vapor within the cyclone is unimpeded, an element of vapor traveling smoothly from a tangential inlet to a central exit.

(*a*) "Standard cyclones" in which the path of the vapor is vertical, the vapor ascending through a central exit. Under this classification are cyclones in which the vertical path is of large diameter in relation to its length, and those in which the vertical-path diameter is small in relation to its length; this ratio being stated by some to be about 0.2 or less.

(*b*) "Webre cyclones" in which the path of the vapor is vertical, the vapor descending through a central exit.

(*c*) "Spiral cyclones" in which the path of the vapor is spiral, the vapor having no vertical velocity in the cyclone.

2 "Complex Cyclones:" In this type the path of the vapor within the cyclone is guided or modified by means of special features or parts of the following categories:

(*a*) Inlets, (*b*) internal parts, (*c*) vapor outlets, (*d*) liquid and dust outlets.

3 "Compound Cyclones:" In this type the vertical path of the vapor within the cyclone is divided into separate vertical paths.

These various types will be outlined briefly. While such performance data as have been published always refer to the separation of dusts from gases and, therefore, may not be analogous to performance data for vapor-liquid streams, the comparison is of interest.

Simple Standard Cyclones of Large Diameter

A simple standard cyclone of large diameter is illustrated in Fig. 1. Reams (49)[3] has tabulated the dimensions of units varying from 4 to 18 ft diam. The American Blower Corporation (5) lists optimum dimensional relations. Dedrick (21) states that the volume of a cyclone should be from $^1/_{20}$ to $^1/_{30}$ that of air handled per minute.

These cyclones are applied most frequently to the separation of coarse dusts and are also used extensively in air-classifying systems, as steam separators. They also appear to be the preferred device for disengaging steam in discharging wood-pulp digesters. Others (8, 14, 61) consider its application for cleaning dust from gases, whereby the gases are sprayed with liquid and

[1] Graduate Student at Columbia University, and Director of Research and Development on By-Products, West Virginia Pulp and Paper Company.

[2] Formerly Associate Professor of Chemical Engineering at Columbia University, now Director of Research and Development, Metal and Thermit Corporation.

Contributed by the Process Industries Division and presented at the Spring Meeting, Atlanta, Ga., March 31–April 3, 1941, of The American Society of Mechanical Engineers.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

the entrained, dust-bearing droplets are thus separated. Mauthe (42) proposes wetted-wall cyclones for cleaning gases.

For separating liquids from vapors, Webre (71) states that this type is comparatively unsuited because of the re-entrainment of liquid at the rim of the outlet tube. Alden (2) limits its effectiveness to dusts above 50 μ. According to the American Blower Corporation (6), the lighter the dust, the lower the air velocity should be; the inlet velocity should not exceed 40 fps. Barth (10) found, on an 8-in. model, that efficiency increases with velocity to a maximum and then decreases. Dedrick's data (21) also indicate that an optimum velocity exists. Drijver (22), on the other hand, reports only an increase with velocity. This was confirmed by Whiton (76) who also showed, testing cyclones 1, 2, and 3 ft in diam, that the smallest diameter gave the greatest dust recovery.

The only generalization which can be deduced from the foregoing references is that simple, large-diameter cyclones are not considered so effective as some of the modified types to be discussed.

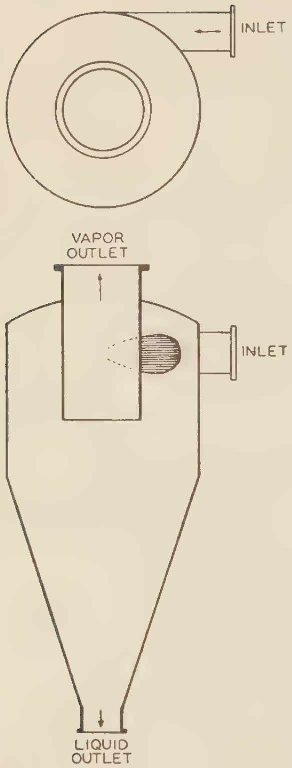


Fig. 1 Standard Cyclone of Large Diameter

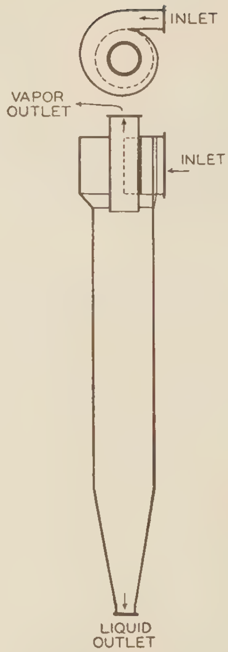


Fig. 2 Standard Cyclone of Small Diameter

Simple Standard Cyclones of Small Diameter

The realization that high velocities improved the efficiency of dust separation led to the development of long narrow cyclones, typified in Fig. 2. They are dimensioned to attain a specific efficiency, increases in capacity being achieved by increasing the number of units. However, this gives rise to complicated distribution problems which, if not adequately solved, minimize the advantages of these cyclones. This type is applied mainly to dusts below 50 μ. Abrasive dusts cause excessive wear. According to Horne and Lissman (33), their vertical height should be at least 6 times their diameter. Lissman (40) discusses units with tubes as small as 4 in. diam. Horne and Lissman have also considered the application of these cyclones for separating entrained droplets from sprays. Various arrangements of distributing systems have been proposed (18, 33, 34, 35, 45, 46, 66, 67). Recently Freeman (28) has applied this type to the separation of dirt particles from thin slurries of wood pulp.

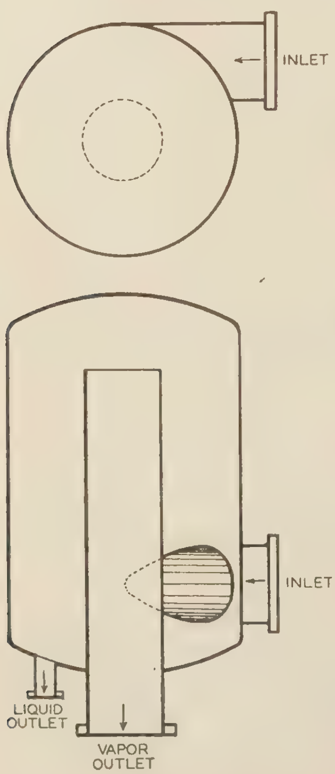


Fig. 3 Webre Cyclone

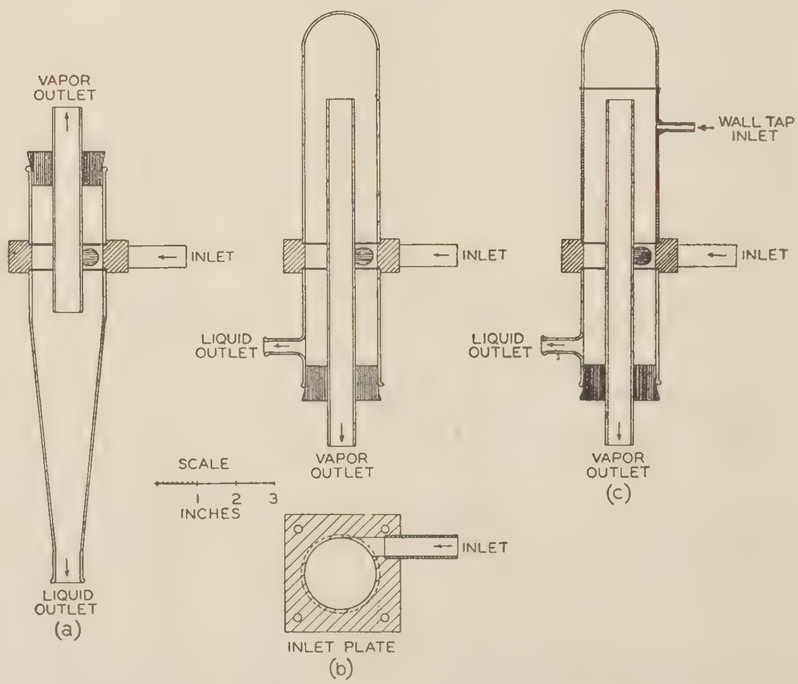


Fig. 4 Standard Cyclone and Two Types of Webre Cyclone
(*a* Standard cyclone, glass model, 1.85 in. diam; *b* Webre cyclone, glass model, 1.85 in. diam; *c* Webre cyclone, 1.85 in. diam, fitted for wall injection.)

No data correlating efficiency and velocity have been published. According to Lissman (40), a collection efficiency exceeding 99 per cent is possible, using 4-in. cyclones, with particles averaging 5 μ. Alden (2) writes that such a cyclone handles 70 cfm.

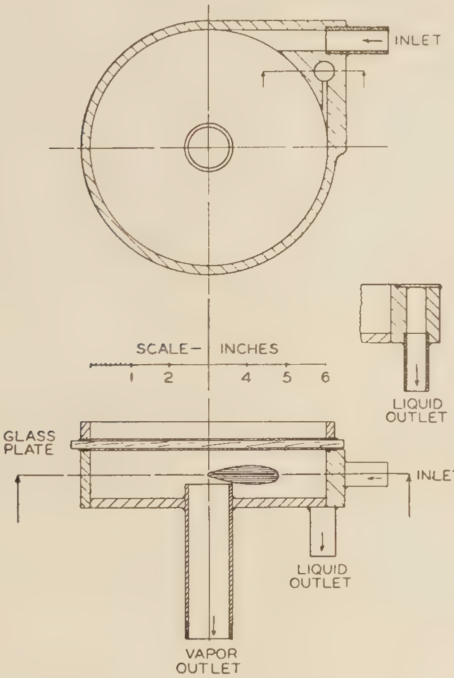


Fig. 5 Spiral Cyclone, 6 In. Diam

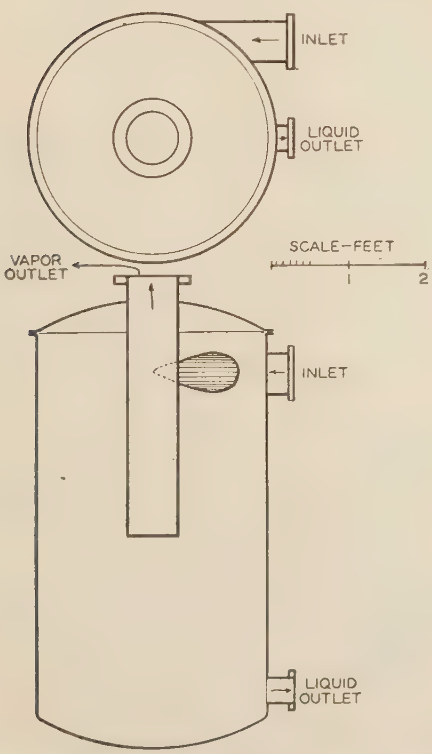


Fig. 6 Standard Cyclone 36 In. Diam, Used for Separating Black Liquor From Steam

Simple Webre Cyclones

Webre cyclones are illustrated in Figs. 3 and 4 (*b*). No data have appeared in the literature on the sizes or dimensions, but units are known to be in operation which vary from 9 in. to over 6 ft in. diam. Webre cyclones have been frequently used as entrainment separators on vacuum evaporators and have also proved effective in diameters below 12 in. for separating liquid from vapor in the flash distillation of oils (27). This type of cyclone appears to be merely an inverted standard cyclone, but Webre and Robinson (71) pointed out that, unlike standard cyclones, liquid does not run along the central outlet pipe to be entrained at the rim.

Simple Spiral Cyclones

Spiral cyclones as, for example, that of Fig. 5, are usually considered to have the advantage of inducing an increasing centrifugal force on particles as they move toward the center outlet, thereby impelling them to move outward against the gas stream. Air grinding classifiers, or micronizers, utilize a somewhat similar design. Frequently, to insure a long spiral path, a spiral baffle is installed, whereby the gas is constrained to move in a spiral path of the desired pitch. Such separators will be considered as complex cyclones.

Complex Cyclones

Many modifications in cyclone design have been proposed to reduce velocity-head losses and to improve efficiencies. While reduced friction losses are comparatively simple to demonstrate, increased efficiency is not so readily ascertained, since variations in particle sizes must be taken into account. Numerous patents and other literature deal with modifications in the four principal structural elements of cyclones as follows:

(*a*) *Inlets.* Various vanes, baffles, or special ports are proposed for obtaining better entrance conditions.

(*b*) *Internal Parts.* Special shapes, baffles, or parts are offered for improving the efficiency of separation.

(*c*) *Vapor Outlets.* Modifications, involving louvers and ports, are devised for aiding the exit of cleaned vapor or gas.

(*d*) *Liquid and Dust Outlets.* Special designs for facilitating the exit of the separated liquid or dust.

Inlets. Alden (2) tabulates dimensional data for cyclones with helical tops used to reduce entrance losses. Capacities from 500 to 40,000 cfm, having diameters from 3 to 14 ft, are given. Deflector plates at the inlet also serve to reduce entrance losses. This has been confirmed by Shepherd and Lapple (54). The use of vanes to initiate the whirling motion is the subject of a number of patents (3, 4, 12, 25, 26, 32, 36, 37, 38, 48, 51, 62, 74). Others (11, 29, 30, 31) prefer louvers for this purpose. Scroll-shaped inlets have also been described (34, 65) Alexander (3, 4) describes a central baffle below the entering zone which forces the fluid material to flow near the wall of the cyclone. Various other inlets have been described (1, 15, 20, 70).

No systematic comparison data have appeared, comparing the efficiencies of any of the wide variety of inlets. Barth (10) demonstrated that a small-diameter inlet gives higher dust recoveries than a large inlet for the same volume of gas handled.

Internal Parts. Barth (10) tested identical cyclones with and without a spiral vane and found the empty cyclone more efficient. Webre (72) considers the spiral-baffled Hughes separator very effective as an entrainment separator in evaporators, but gives no data. The use of spiral vanes is also the subject of patents (11, 24, 44, 67). Drijver (22) tested vanes placed against the shell of a cyclone operated as a dust separator. With the

vanes pitched into the gas stream, he obtained a 35 per cent dust recovery. Pitched so as not to oppose the gas stream, the recovery was 55 per cent. Removed altogether, the recovery rose to 75 per cent. He attributed their detrimental action to re-entrainment. Alden states (2) that reversing vanes below the outlet pipe increased the resistance to a total of 1 to 2 velocity heads but that they also increased the separating efficiency for small particles. A disk or cone, centered below the outlet pipe, is also frequently seen. Other patents on this type of improvement have been granted (12, 15, 18, 20, 25, 41, 44, 51, 55, 56, 64, 66, 67, 68).

Vapor Outlets. Schaak (52) found that straightening vanes below the exit tube reduced the velocity-head losses approximately 50 per cent. Shepherd and Lapple (54) confirm this. No suggestion of its effect on separating efficiency is offered. Patents on the use of straightening vanes have been secured (36, 37, 48). Prockat (47), experimenting with dust, compared a cyclone containing a widely flared exit tube with one having no internal exit tube and found the latter much more efficient.

Many designs show louvers inserted into the exit tube. In the Van Tongeren cyclone (65), the exit tube is replaced by a ring of reversing louvers, whereby the gas reverses its flow. The length of the outlet tube does not appear to be important, according to Barth (10). As might be expected, supports and other obstructions were found to decrease the efficiency of separation.

Liquid and Dust Outlets. A number of improvements over the common hopper-bottom dust outlet have been devised. In some the dust is withdrawn as an annular ring near the wall by using baffles (3, 4, 63, 77). The use of vertical slots for skimming off liquid or dust has also been considered in patents (4, 13, 65, 68, 70), as have series of perforations (64, 66, 67).

Compound Cyclones

Increased efficiency is claimed for the compound cyclone. Re-entrainment, especially of the fine particles near the wall, is minimized by skimming off a layer of gas near the wall and transferring this dust-rich layer to a second cyclone. The Van Tongeren cyclone of this design is claimed to give 80 per cent efficiency with 10 μ particles (16, 65). Stebbins (59) and O'Mara (46) show similar arrangements. According to Alden (2), identical cyclones in series are not worth while.

Factors Influencing the Investigation of Cyclone Performance

When cyclones are used for separating liquids from vapors, the system consists of the following:

1 A vapor-liquid mixture entering the cyclone.

2 A cleaned-vapor effluent, containing a residual portion of unseparated or re-entrained liquid, the entrainment.

3 A liquid effluent, consisting of the separated liquid.

The entrainment is the main criterion of the performance of a cyclone. For a specific cyclone, handling a specific vapor-liquid stream, there should be some relation among the vapor velocity, the amount of liquid, and the particle-size distribution of any liquid in droplet form.

In discussing the action of cyclones, writers (78, 79, 80, 81) have generally reasoned that increasing the velocity increased the effectiveness of separation. However, it is also commonly recognized that a cyclone can be overloaded. Any adverse effect of velocity on the effectiveness of separation has always been tacitly ascribed to secondary effects arising from supposed structural defects in the particular cyclone. In spite of the many varied designs of special inlets, baffles, and outlets, which have been proposed, there appears to be an increasing agreement among experienced designers that baffles and other complicating features are of doubtful effectiveness.

In cyclones applied to the separation of dusts, the particle-size distribution is an important factor in the effectiveness of separation, especially when any considerable portion comprises particles below 50 μ. In separating vapor-liquid mixtures from stills, evaporators, or pressure vessels, the size of droplets is not always so important. In practice, cyclones are known to remove most of the entrained liquid. Consequently, it would seem likely that most of the liquid in such vapor-liquid streams occurs either in bulk or as droplets coarse enough to be separable by centrifugal force. The presence of any considerable amount of liquid in the form of droplets of a few μ diameter is also unlikely, in view of the large amount of energy required for disintegration. This would suggest that the presence of small droplets is not necessarily a dominant factor in the performance of cyclones separating considerable liquid.

Experimental Procedure

To facilitate the correlation of data, all quantities of liquid and vapor are expressed in terms of weight velocities referred to the cross section of the inlet tube as follows:

L_v = pounds of liquid leaving per minute in effluent vapor per square foot of inlet cross section, lb per min per sq ft

V = pounds of vapor entering per minute per square foot of inlet cross section, lb per min per sq ft

L = pounds of liquid entering per minute per square foot of inlet cross section, lb per min per sq ft

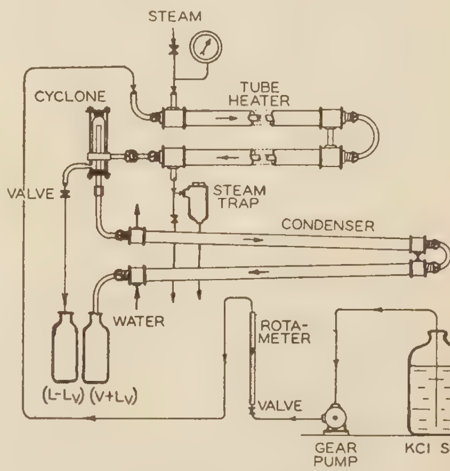


Fig. 7 General Arrangement of Apparatus for Entrainment Tests, Using Heater Tubes

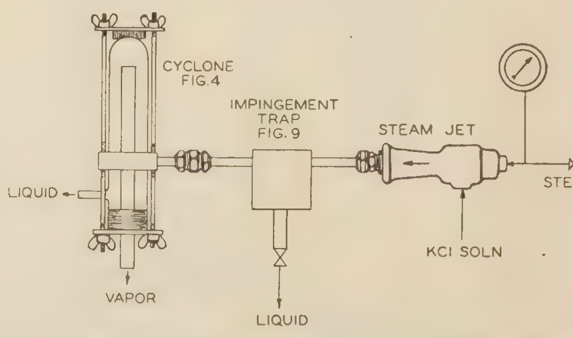


Fig. 8 General Arrangement of Apparatus for Entrainment Tests, Using a Steam Jet and Impingement Trap

Accordingly, the vapor-liquid mixture entering the cyclone is $(V + L)$; the cleaned vapor effluent containing residual liquid is $(V + L_v)$; the liquid effluent is $(L - L_v)$.

The entrainment in cyclones could be defined in a number of ways. Thus, viewed as an impurity in the vapor, the entrainment may be considered to be the ratio of (L_v/V) or, viewed as a loss of liquid, it may be (L_v/L). For the purpose of this research, the entrainment is defined as L_v, the pounds of liquid leaving per minute with the effluent vapor per square foot of inlet cross section.

The operating and design variables of various cyclones were investigated in 15 series of experiments. The cyclones used are illustrated in Figs. 4, 5, and 6. A jacketed-tube heater, Fig. 7, whereby water or dilute potassium-chloride solution could be partially vaporized, was used for generating continuous streams of steam-water mixtures. Fig. 8 illustrates the application of steam jets to supply such mixtures.

The condensed vapor effluent $(V + L_v)$ and the separated

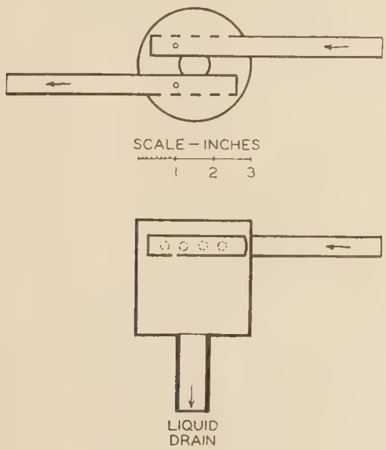


Fig. 9 Impingement Trap Inserted in Vapor-Liquid Line Before the Cyclone

liquid effluent $(L - L_v)$ were collected for one or more minutes and measured in suitable graduated cylinders. Both effluents were analyzed for chloride ion by titrating an aliquot with approximately 0.1 molar silver-nitrate solution, using sodium chromate as an indicator. Where the vapor effluent was too dilute to secure distinct end points, or the titer was too small, the potassium-chloride concentration was estimated by means of a Leeds and Northrup portable electrolytic resistance indicator and dip-type conductivity cells. Calibration curves for the two cells used were made from measurements using various dilutions of 0.1 molar potassium-chloride solution. Temperature corrections were made by cooling or warming samples to 25 ± 5 C and then noting the temperature to 0.5 deg, when reading the resistance meter. All resistances were converted to a 25 C basis by correcting readings 2 per cent per deg deviation. Wherever the concentrations permitted, the conductivity method was checked by the titration method. The results invariably checked within several per cent.

At the operating rates encountered in these experiments, errors due to vapor condensation in the cyclone or liquid flashing at the cyclone inlet could safely be ignored, so that the separated liquid, $(L - L_v)$, and the entrained liquid, L_v, were considered identical in potassium-chloride concentration. Since the potassium-chloride concentration in $(L - L_v)$ and in $(V + L_v)$ was determined, L_v could be computed.

The experiments are ordered chronologically, the upper-case letters referring to a specific arrangement of equipment and the numbers to individual experiments in which V or L was varied.

Some data were collected on the pressure drops in the cyclones but omitted from this paper since it did not appear to be pertinent. Efforts were also made to gage the velocity head at various points within the 1.85-inch Webre cyclone by inserting shaped copper capillary tubes but this cyclone was so small that excessive refinements in such a technique appeared necessary for consistent readings.

Entrainment Correlations in 1.85-In. Webre Cyclone

Using the 1.85-in. Webre cyclone of Fig. 4 (*b*) and the jacketed-tube vapor generator, shown in Fig. 7, a very definite relation among the entrainment L_v, the vapor velocity V, and the liquid velocity L was found. The simplest correlation appears to be

$$L_v\left(\frac{V}{L}\right)^2 = c_1V + b_1 \quad ..[1] \quad \text{or} \quad \text{Log}\left[L_v\left(\frac{V}{L}\right)^2\right] = b_2 + c_2V \quad ..[2]$$

where a, b, and c are constants.

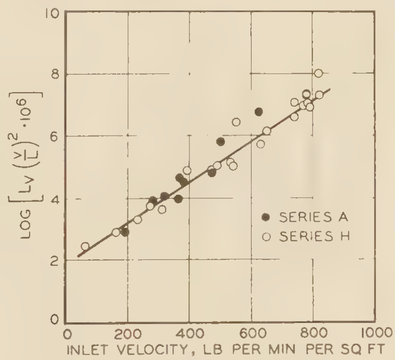


Fig. 10 Entrainment Relation in a 1.85-In. Webre Cyclone
(Series A using 24-in. heater tubes; series H using 48-in. heater tubes, arranged as in Fig. 7. Data from Table 1.)

The results of 31 experiments in two series, covering wide variations in L and V, are summarized in Table 1 and the functional relation of L_v, V, and L is illustrated in Fig. 10. For the experiments of series A, two 24-in. heater tubes, 0.58 in. inside diam were used. In series H these were replaced with two 48-in.

TABLE 1 ENTRAINMENT CORRELATIONS IN A 1.85-IN. WEBRE CYCLONE

Experiment no.	Vapor velocity, lb per min per sq ft V	Liquid velocity, lb per min per sq ft L	Entrainment, lb per min per sq ft L_v	Entrainment function, $\log\left[L_v\left(\frac{V}{L}\right)^2 \times 10^6\right]$	Remarks
A- 1	620	960	13.0	6.73	KCl solution metered into two 24-in. heater tubes
A- 2	360	1310	0.13	3.98	
A- 3	500	1100	2.9	5.80	
A- 4	370	1280	0.53	4.66	
A- 5	380	1260	0.37	4.53	
A- 6	280	1360	0.18	3.90	
A- 7	320	1330	0.19	4.04	
A- 8	190	1560	0.052	2.86	
A- 9	470	1110	0.36	4.80	
H- 1	390	780	0.331	4.91	KCl solution metered into two 48-in. heater tubes filled with jack chain
H- 2	530	610	0.19	5.16	
H- 3	790	140	0.28	6.96	
H- 4	740	570	8.4	7.16	
H- 5	780	270	3.01	7.41	
H- 6	820	230	9.2	8.07	
H- 7	780	440	7.3	7.36	
H- 8	550	900	8.8	6.51	
H- 9	820	110	0.42	7.35	
H-10	630	1370	2.78	5.76	
H-11	650	1310	6.2	6.19	
H-12	540	2410	2.26	5.06	
H-13	770	130	0.31	7.03	
H-14	470	2840	2.9	4.89	
H-15	780	90	0.158	7.08	
H-16	310	430	0.008	3.64	
H-17	270	750	0.044	3.74	
H-18	490	530	0.139	5.08	
H-19	230	830	0.028	3.32	
H-20	160	870	0.00024	2.89	
H-21	740	150	0.173	6.63	
H-22	60	540	0.0254	2.47	

tubes of the same diameter but filled with brass jack chain to insure intimate mixture.

Entrainment Correlations in a 36-In. Standard Cyclone

A standard cyclone 36 in. diam, Fig. 6, was tested at the Tyrone, Pa., mill of the West Virginia Pulp and Paper Company, where it served to separate black liquor from steam. This cyclone was so installed that, while the effluent vapor $(V + L_v)$ could be sampled and measured by means of a pitot tube in the piping from the cyclone to a condenser, the effluent-separated-liquid piping was accessible only for sampling the liquid $(L - L_v)$.

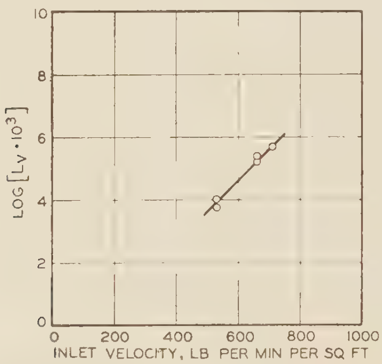


Fig. 11 Entrainment Relation in a 36-In. Standard Cyclone, Separating Black Liquor From Steam
(Data from Table 2.)

However, it was known that the ratio V/L was nearly constant, so that a simplified form of Equation [2] should apply, namely

$$\text{Log } L_v = b_3 + c_3V \qquad [3]$$

The data of Table 2 and the corresponding Fig. 11, would indicate such a relation. While this does not demonstrate that, if the ratio V/L had varied, Equation [2] would have applied, nevertheless, it serves to show that the entrainment increases with the inlet-weight velocity V in this cyclone also.

TABLE 2 ENTRAINMENT OF BLACK LIQUOR IN A 36-IN. STANDARD CYCLONE

Experiment no.	Vapor velocity, lb per min per sq ft V	Entrainment, lb per min per sq ft L_v	Entrainment function log $[L_v \times 10^3]$
P-1	530	6	3.78
P-2	530	11	4.04
P-3	660	260	5.41
P-4	660	170	5.23
P-5	710	530	5.72

The Influence of Droplet Size on Entrainment

To increase the proportion of liquid entering the cyclone in droplet form as compared to liquid of greater bulk, an impingement trap Fig. 9 was inserted in the vapor-liquid line before the cyclone. Most of the liquid present as large drops or in bulk was thus drained from the bottom of the trap. In some of the experiments, the tube heaters were replaced by two sizes of steam-jet injectors, Fig. 8.

The results of the four series of experiments so made are summarized in Table 3 and illustrated in Fig. 12. It will be seen that in all cases the entrainment function, Equation [2], furnished a definite correlation which may be compared with the results of series A and H in Fig. 10. The constants b and c, corresponding to the intercept and slope of the lines of Fig. 12, are thus seen to be influenced by the nature of the liquid, presumably by the particle-size distribution of the liquid in the liquid-vapor stream.

TABLE 3 INFLUENCE OF DROPLET SIZE ON ENTRAINMENT

Experiment no.	Vapor velocity, lb per min per sq ft V	Liquid velocity, lb per min per sq ft L	Entrainment, lb per min per sq ft L_v	Entrainment function log $\left[L_v\left(\frac{V}{L}\right)^2 \times 10^6\right]$	Steam-jet size, in.	Steam-jet pressure psi gage
D2a	530	21	1.09	8.82	3/8	43
b	230	34	0.65	7.49	3/8	43
D3a	550	29	1.02	8.57	3/8	42
b	240	32	0.63	7.56	3/8	42
E1a	700	89	0.15	6.97	3/4	11
b	470	121	0.094	6.16	3/4	11
E2a	790	118	0.26	7.06	3/4	13
b	380	126	0.18	6.21	3/4	13
E3a	870	134	0.40	7.24	3/4	17
b	550	173	0.21	6.34	3/4	17
F1a	520	31	0.92	8.41	3/8	43
b	240	25	0.58	7.75	3/8	43
F2a	130	7.8	3.0	8.92	3/8	20
b	210	9.2	4.6	9.37	3/8	20
c	290	14	8.6	9.59	3/8	20
F3a	510	22	0.91	8.70	3/8	40
b	420	25	0.76	8.34	3/8	40
c	220	21	0.66	7.89	3/8	40
F4a	290	8.0	4.1	9.73	3/8	20
b	120	8.1	3.8	8.97	3/8	20
G2a	640	155	0.28	6.69	two 48-in. tube heaters	
b	310	100	0.150	6.18		
G3a	720	200	0.45	6.76		
b	360	135	0.16	6.07		
G4a	710	260	0.44	6.53		
b	380	160	0.155	5.93		
G5a	830	140	0.32	7.08		
b	360	100	0.135	6.22		

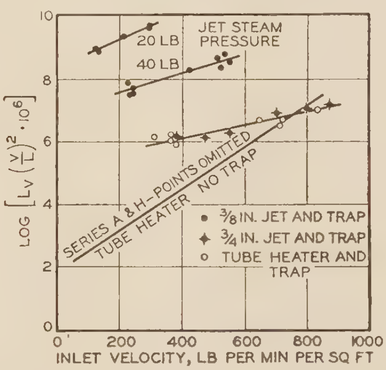


Fig. 12 Influence of Droplet Size on Entrainment, Using Webre Cyclone
(Data from Table 3.)

Thus, the small steam jet displaces the function to the greatest degree.

Design Variables in 1.85-In. Cyclones

Comparison of Webre and Standard Cyclones. A series of comparison tests was made, changing from Webre to standard cyclone by changing the glass parts of Fig. 4 (*b*) to those of Fig. 4 (*a*). The metal inlet plate was remachined to furnish a 0.25-in. tangential inlet in place of the 0.50-in. inlet used in previous experiments. Otherwise, the arrangement of apparatus was that of Fig. 7, no trap being used. The results are recorded in Table 4 and shown in Fig. 13.

The difference in operation between the cyclones was quite noticeable. Where, in the Webre cyclone, the outer wall of the exit tube remained dry and liquid spread over the shell only, in the standard cyclone, considerable liquid ran down the outer wall of the exit tube to be entrained in the vapor. Varying the height of the shell above the inlet plate from 0 to 2 in. did not appear to decrease the movement of liquid onto the exit-tube wall. These results confirm Webre's contention that a central exit tube in the top end of a cyclone aids entrainment.

Inlet Sizes of Webre Cyclone. The effect of varying the inlet size from 0.50 to 0.25 in. diam may be noted by comparing, in Fig. 13, the data of series L for a 0.25-in. inlet, with the earlier series H, where a 0.50-in. inlet was used. Increasing the inlet velocity displaced the entrainment function, Equation [2].

TABLE 4 DESIGN VARIABLES IN 1.85-IN. CYCLONES

Experiment no.	Vapor velocity, lb per min per sq ft V	Liquid velocity, lb per min per sq ft L	Entrainment, lb per min per sq ft L_v	Entrainment function $\log\left[L_v\left(\frac{L}{V}\right)^2 \times 10^6\right]$	Cyclone
L-1	1320	8500	102	6.39	Webre, 0.25-in. inlet
L-2	1510	2040	102	7.75	
L-3	740	3160	0.17	4.00	
N-1	2030	1950	295	8.50	Standard, 0.25-in. inlet
N-2	380	1340	460	7.57	
N-3	940	1490	325	8.11	
M-1	1120	1840	0.74	5.43	Webre, 0.25-in. inlet

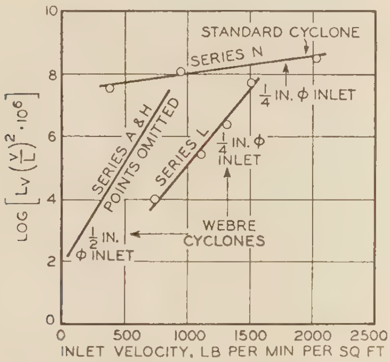


FIG. 13 EFFECT OF DESIGN VARIABLES ON ENTRAINMENT RELATION, USING 1.85-IN. CYCLONES
(Data from Table 4.)

THE CYCLONE AS THE SOURCE OF ENTRAINMENT

The entrainment relation expressed by Equation [2] has survived experiments involving wide variations in V and L and also some design variations. If then this function holds so generally, it would seem likely that it would be due to actions within the cyclone rather than to the states and rates of the vapor-liquid stream.

To investigate this consideration, a tap was machined, Fig. 4 (*c*), into the side wall of the 1.85-in. Webre cyclone, whereby a small stream of potassium-chloride solution could be pumped into the cyclone. By feeding tap water into the two 48-in. vaporizing tubes, all liquid entering the cyclone from the vaporizer was, within the limits of the authors' precision, free of chloride ion. Consequently, entrainment due to droplets of liquid formed outside the cyclone and contained in the inlet stream would not be detected by analyses of samples of the effluent vapor. On the other hand, liquid on the wall of the cyclone, being mixed with the potassium-chloride solution pumped in at the wall tap, would, if entrained, be measured by a corresponding amount of chloride ion in the effluent vapor ($V + Lv$).

The results of these experiments, series I, are listed in Table 5. In Fig. 14 the entrainment relation is compared with that for series A and H. It will be seen that curve I is somewhat displaced. This and the points off the curve were judged to be due to the incomplete mixing of the water on the wall with the potassium-chloride solution fed.

A confirmatory series of experiments was therefore made, using the same apparatus except that the 0.50-in. inlet was replaced by a 0.25-in. inlet. This smaller inlet resulted in higher inlet velocities and, therefore, better mixing of the wall liquids. The results of series L, already listed in Table 4, were found, as noted, to be similar to those of series A and H, potassium chloride being fed through the heater tubes. In another experiment M-1, the procedure of feeding potassium-chloride solution through the cyclone-wall tap was followed. Fig. 15 shows how this result conforms to those of series L. It would appear that the entrainment is due to actions within the cyclone in addition to the properties of V and L.

TABLE 5 ENTRAINMENT FROM LIQUID ON CYCLONE WALL

Experiment no.	Vapor velocity, lb per min per sq ft V	Liquid velocity, lb per min per sq ft L	Entrainment, lb per min per sq ft L_v	Entrainment function $\log\left[L_v\left(\frac{V}{L}\right)^2 \times 10^6\right]$
I-1	720	250	5.7	7.69
I-2	740	250	2.7	7.38
I-3	720	250	2.1	7.24
I-4	710	250	2.4	7.20
I-5	550	1460	14.5	6.32
I-6	570	1410	17.8	6.47
I-7	560	1390	31.5	6.71
I-8	360	2970	56.5	5.91
I-9	270	3900	0.29	3.13

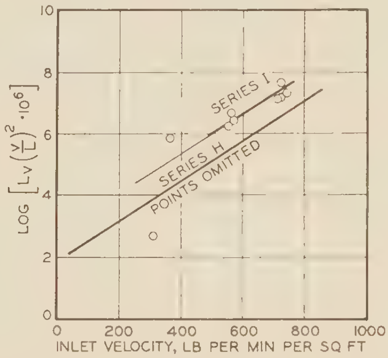


FIG. 14 ENTRAINMENT FROM LIQUID ON CYCLONE WALL; ENTRAINMENT RELATION USING 0.50-IN. INLET
(Data from Table 5.)

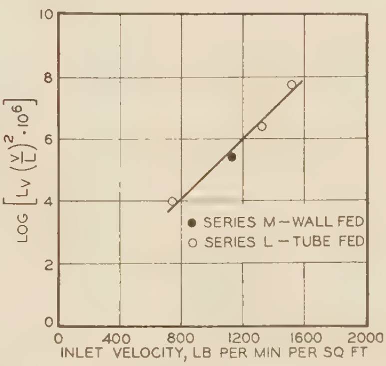


FIG. 15 ENTRAINMENT FROM LIQUID ON CYCLONE WALL; ENTRAINMENT RELATION USING 0.25-IN. INLET
(Data from Table 4.)

THE WALL CREEP OF LIQUID AS A SOURCE OF ENTRAINMENT

In all the cyclones which were tested by the authors, the entrainment is an increasing function of the vapor velocity. This function persists even when the entrainment due to droplets of L is discounted, as in series I and M, or when the proportion of L in droplet form is increased as in series D to G.

If the entrainment is to be attributed to some action occurring within the cyclone, then some explanation or description of the mechanism for this action is needed. Most of the liquid entering flows onto the wall of the cyclone, forming a turbulent, continuous layer which can be seen to move in the direction of the vapor with which it is in contact. It is obvious that the vapor entering a cyclone pursues a spiral or vortical path to a central outlet. Using standard cyclones Shepherd and Lapple (54) and others have shown that the flow consists of an outer downward vortex and an inner upward vortex. Secondary actions have also been

reported, notably by Van Tongeren (65) and Wellman (73). The former states, without giving data, the existence of a so-called double eddy current of gas which serves to transport dust inward at the extremities of a cyclone. Shepherd and Lapple were unable to detect the presence of such a current, nor did observations made during the authors' experiments suggest the action of such a current.

Perhaps a simpler explanation for entrainment actions within a cyclone is conceivable, at least in the separation of liquid from vapor. If, as most writers indicate, and as has been observed invariably in these experiments, the vapor entering a cyclone pursues some form of spiral or vortex, then the vapor velocity must at all places have a component radially inward. Liquid on the walls must then be induced to creep along surfaces toward the center exit.

TABLE 6 ENTRAINMENT CORRELATIONS IN A 6-IN. SPIRAL CYCLONE

Experiment no. (series O)	Vapor velocity, lb per min per sq ft V	Liquid velocity, lb per min per sq ft L	Entrainment, lb per min per sq ft L_v	Entrainment relation log $[L_v \times 10^3]$
1	640	310	68	4.83
2	290	530	105	5.02
3	121	690	0.76	2.88
4	97	650	0.26	2.41
5	153	1320	0.65	2.81
6	360	980	97	4.99
7	640	600	66	4.82
8	650	450	63	4.80
9	610	1280	71	4.85
10	600	1510	71	4.85
11	580	1550	73	4.86
12	510	1300	81	4.91
13	490	1270	81	4.91
14	490	1300	83	4.92
15	370	1390	105	5.02
16	360	1340	107	5.03
17	360	1380	111	5.05
18	121	1640	8.6	3.93
19	108	1700	10.3	4.02
20	162	1410	0.61	2.79
21	270	1130	123	5.09
22	230	1160	134	5.13
23	210	1360	129	5.11
24	210	1330	131	5.12
25	137	1280	0.5	2.68
26	162	1230	0.075	1.87
27	157	1250	0.81	2.91
28	111	1280	6.3	3.80
29	118	1250	7.3	3.86
30	400	3080	123	5.09
31	360	3400	131	5.12
32	190	2970	23.8	4.38
33	200	2970	27.5	4.44

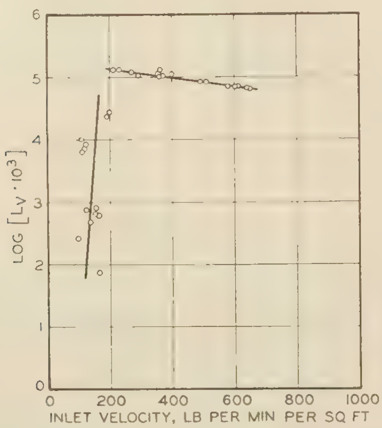


FIG. 16 ENTRAINMENT RELATION IN 6-IN. SPIRAL CYCLONE
(Data from Table 6.)

To enable better visual observations, in addition to the collection of data, a spiral cyclone 6 in. diam was made, fitted with a Pyrex plate-glass top, assembled as illustrated in Fig. 5. This cyclone was operated with the same vaporizing tubes and other equipment shown in Fig. 7, as was used in series H. In addition to the usual measurements for determining L_v, V, and L, the plate-glass top afforded visual evidence of the entrainment action.

The results of 33 experiments are summarized in Table 6. The correlation previously noted, Equation [2], did not hold for this cyclone. The data collected in this series of experiments would make it appear that there is a critical velocity below which L_v increases rapidly with V, and above which L_v decreases slowly with V. Variations in L do not appear to influence the entrainment. The functions appear to be of the form

$$\text{Log } L_v = b_4 + c_4 V \qquad [4]$$

These relations are illustrated in Fig. 16. Surprisingly definite patterns of liquid flow were visible on the glass plate when the cyclone was in operation. At vapor velocities below the critical of 200 lb per min per sq ft of inlet section the pattern was unstable. Liquid would tend to remain near the wall of the cyclone. Some of the liquid on the glass plate would tend to move along the glass plate to the center, but reactions would develop periodically whereby centrifugal forces would sling the liquid back in waves. As the vapor velocity was increased toward the critical, liquid began to move toward the center with greater velocity and the beginnings of the much more stable high-velocity pattern could be discerned.

At vapor velocities above the critical, a definite geometric pattern was invariably established. The principal features of this pattern are easy to discern. Most prominent is the smooth, spiral streaming of liquid from the wall of the cyclone to the center. By focusing a camera on the interior surface of the glass cyclone cover plate, it was possible to high-light some of the streaming liquid and to photograph the spiral flow. Such a photograph, reproduced in Fig. 17, was taken with the cyclone operating at an inlet-vapor velocity of 780 lb per min per sq ft. Term L was nearly zero in order to reduce the liquid flow to a few well-defined streams. Two of the curves from opposite sides of the center of the cyclone were calipered from the photograph and the data, corrected for the photographic reduction, caliper $\times 1.29$ = actual size, is given in Table 7. It is of interest to note that these curves are logarithmic spirals

$$r = 2.79^{\theta} \qquad [5]$$

where

r = radius vector, in. $\times$ 100
θ = vector angle, in radians

Fig. 18 shows the small deviation between the measured points and the logarithmic spiral of Equation [5].

A second feature of the liquid flow pattern is the presence of an inner spinning ring of liquid. The diameter and thickness of this ring vary inversely with the vapor velocity. This ring becomes unstable when, at low values of V, the diameter of the ring exceeds the inside diameter of the vapor outlet pipe, 1.06 in. As V increases, the ring becomes thinner and decreases in diameter until, at the highest values reached, about V = 800 lb per min per sq ft, the ring reduces to about 0.1 in. diam. At intermediate velocities, when the ring is about 0.5 in. diam, an inner spiral of liquid flowing outward from the center point of the plate and to the inner ring is clearly visible. This inner spiral of liquid rotates in a direction opposite to the large spiral and appears to sling back onto the inner ring any liquid which gets any nearer to the center point of the cyclone than the inner ring.

It appears, therefore, that the spinning ring marks the equilibrium of forces along the glass plate between the drag of the radial component of the vapor velocity and the centrifugal force of the liquid spinning at the center. Liquid can be seen to drop into the vapor outlet pipe from the spinning ring and also from the center of the inner spiral of liquid. These observations are illustrated

TABLE 7 SPIRAL-FLOW DATA CORRECTED FOR PHOTOGRAPHIC REDUCTION OF FIG. 17

Curve A		Curve B	
Degrees from reference axis	Radius, in. × 100	Degrees from reference axis	Radius, in. × 100
0	270	240	135
30	185	270	75
60	120	300	40
90	64	330	29
120	33	360	22
150	22	...	...
180	14	...	...

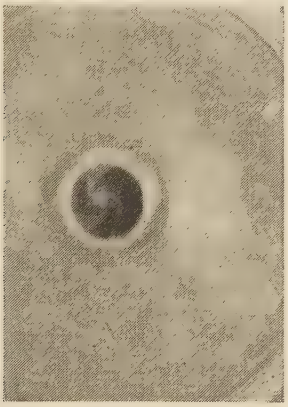


Fig. 17 Wall Creep in Spiral Cyclone

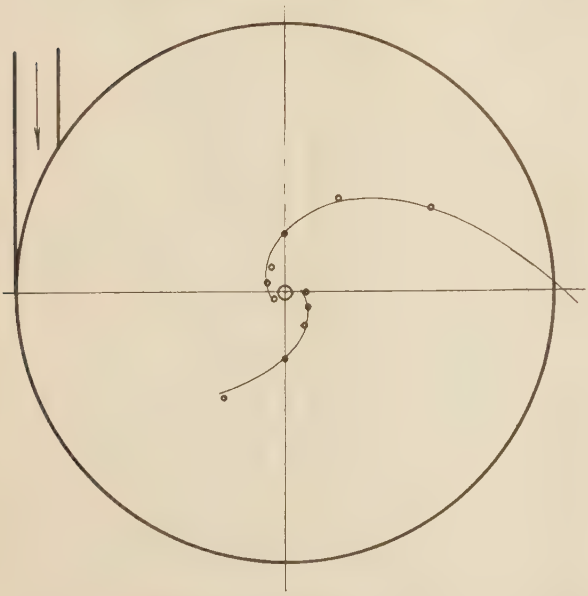


Fig. 18 Calipered Points Falling on Curve $r = 2.79^{\theta}$, Measured From Photograph of 6-In. Spiral Cyclone

in Fig. 19, showing plan and diametral section views of the cyclone. It is not difficult to observe an analogous wall creep in the case of the 1.85-in. Webre cyclone used in previous experiments. While the shape of this cyclone does not lend itself to photography or measurement, a sketch, Fig. 20, shows in section the typical liquid distribution. It will be seen that, in so far as the curved shell permits, the pattern of liquid distribution conforms to that for the spiral cyclone.

Discussion of Experimental Results

The logarithmic spiral of Equation [5] describes the path of the liquid which is being entrained. Term θ, the angular velocity, must be a function of the inlet-vapor velocity. Radius vector r, must be a function of the radial velocity of the vapor as well as of the liquid being entrained.

The similarity of the two generalized functions of Equations [4] and [5]

$$L_v = c^{(V+b)} \quad \text{[6]}$$

and

$$r = n^{(\theta + m)} \quad \text{[7]}$$

suggests an analogy between the results of the experiments of series I, L, and M and the present observations. The parallelism suggests that the radial component of the vapor velocity is the force which induces this wall creep.

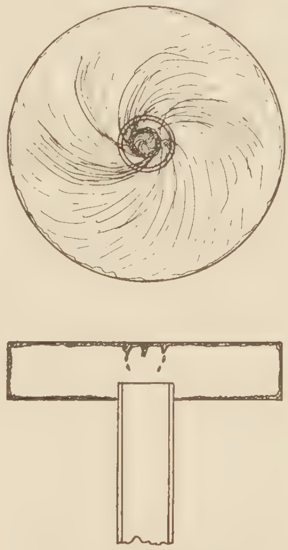


Fig. 19 Wall Creep in 6-In. Spiral Cyclone

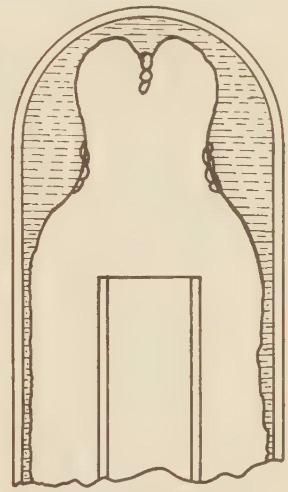


Fig. 20 Wall Creep in 1.85-In. Webre Cyclone

The significance of the factor $(V/L)^2$ in the more complicated entrainment function of Equations [1] and [2] is not clear. However, it may be noted that Equation [1] rewritten as

$$L_v = (L/V)^2 \times c^{(V+b)} \quad \text{[8]}$$

suggests that the factor $(L/V)^2$ may be associated with the energy required to lift liquid from the inlet level to the level where it may creep inward. Such a factor would not occur in the 6-in. spiral cyclone where no lift is required. In confirmation of this, it may be noted that the entrainment L_v is much greater for the spiral cyclone than for the Webre cyclone.

If such a lift is the explanation for the presence of this factor in Equation [1], then it would follow that the V in this factor is not the inlet velocity but rather the vertical component of the internal velocity. This could be investigated further by testing similar cyclones of varying dimensions. The results so far suggest the generalized form

$$\text{Log}\left[L_v\left(\frac{V}{L}\right)^a\right] = b + cV \quad \text{[9]}$$

where $a = 0$ in the case of the spiral cyclone and $a = 2$ in the cases of the other cyclones tested.

Conclusions

1 The entrainment, or liquid remaining in the cleaned vapor of cyclones, separating liquid from vapor, is a function of the liquid- and vapor-inlet weight velocities.

2 On the basis of this investigation, it appears that the entrainment is related to the liquid and vapor weight velocities by the function

$$\text{Log}\left[L_v\left(\frac{V}{L}\right)^a\right] = b + cV$$

where Lv is the entrainment, V is the vapor velocity, and L is the liquid velocity, all in the units of pounds per minute per square foot of inlet-duct cross section. Terms a, b, and c are constants associated with the type of cyclone and the condition of the liquid in the vapor-liquid stream.

3 Under all the operating conditions encountered in the cyclones tested, the constant a is zero in the case of a spiral cyclone and equals 2 in the case of a Webre cyclone.

4 The constants b and c are affected by variations in droplet size and also by the type of cyclone used.

5 In these experiments, at velocities below 1500 lb per min per sq ft, the entrainment with a Webre cyclone was lower than that with a similar standard cyclone.

6 From these experiments, a spiral cyclone appears to be less effective than a Webre cyclone.

7 The creep of liquid along the surfaces of a cyclone, leading to the central vapor-exit tube, was found to be a significant factor in the entrainment encountered in the cyclones used in this investigation.

8 In the case of a spiral cyclone, the path of liquid moving toward the central vapor-exit tube appears to be a logarithmic spiral.

9 The creep of liquid is induced by the inward radial component of the vapor velocity.

BIBLIOGRAPHY

1 U. S. Patent No. 1,825,990; R. C. Aland.

2 "Cyclones and Filters," by J. L. Alden, *Heating and Ventilating*, vol. 35, July, 1938, pp. 48–53.

3 U. S. Patent No. 1,960,887; W. Alexander.

4 U. S. Patent No. 1,898,608; W. Alexander.

5 "Air Conditioning and Engineering," American Blower Corporation, Detroit, Mich., 1935, p. 147.

6 "Air Conditioning and Engineering," American Blower Corporation, Detroit, Mich., 1935, p. 315.

7 U. S. Patent No. 2,032,827; N. H. Andrews.

8 U. S. Patent No. 2,155,853; A. W. Anthony, Jr.

9 "Heat Transfer and Evaporation," by W. L. Badger, Chemical Catalog Company, New York, N. Y., 1926, pp. 212–214.

10 "Die Anwendung des Modellversuches in der Staubtechnik," by W. Barth, *Rauch und Staub*, vol. 22, 1932, pp. 93–98.

11 U. S. Patent No. 1,950,285; R. Battle, and F. I. Ross.

12 U. S. Patent No. 1,734,030; A. C. Bennett.

13 U. S. Patent No. 1,953,948; T. C. Bieth.

14 U. S. Patent No. 1,979,189; D. W. Bowers.

15 U. S. Patent No. 1,774,588; C. P. Brockway.

16 "Fly-Ash Prevention," by S. Brown, TAPPI, Technical Association Papers, series 21, 1938, pp. 207–209.

17 U. S. Patent No. 1,780,641; S. N. Chew.

18 U. S. Patent No. 1,970,077; T. B. Collins.

19 U. S. Patent No. 1,756,909; L. V. Cram.

20 U. S. Patent No. 1,708,122; H. R. Davies.

21 "Cyclone Dust Collector Experiments," by B. W. Dedrick, *National and American Miller*, vol. 60, 1932, pp. 11, 12, 80, 82.

22 "Untersuchungen zur Verbesserung des Auffangwertes eines Zyklons für Hochofenstaub," by A. Drijver, *Wärme*, vol. 60, 1937, pp. 333–339.

23 U. S. Patent No. 1,784,664; F. Eickholt.

24 U. S. Patent No. 1,764,250; E. Falkenthal.

25 U. S. Patent No. 1,899,498; J. H. Fedeler.

26 U. S. Patent No. 1,824,713; J. P. Fisher and G. F. Downs.

27 U. S. Patent No. 2,143,344; E. M. Frankel and A. Pollak.

28 "The Vortrap," by H. Freeman, TAPPI, Technical Association Papers, series 21, 1938, pp. 225–228; "Separation of Solid Particles From Fluids," December, 1937, U. S. Patent No. 2,102,525.

29 U. S. Patent No. 1,684,022; C. G. Hawley.

30 U. S. Patent No. 1,883,908; C. G. Hawley.

31 U. S. Patent No. 2,004,467; C. G. Hawley.

32 U. S. Patent No. 2,004,468; C. G. Hawley.

33 U. S. Patent No. 1,886,548; G. H. Horne and M. A. Lissman.

34 U. S. Patent No. 1,930,806; G. H. Horne and M. A. Lissman.

35 U. S. Patent No. 1,973,764; G. H. Horne and M. A. Lissman.

36 U. S. Patent No. 1,746,218; H. G. Kamrath.

37 U. S. Patent No. 1,757,432; H. G. Kamrath.

38 U. S. Patent No. 1,761,923; H. G. Kamrath.

39 U. S. Patent No. 2,006,244; J. F. Kopsa.

40 "An Analysis of Mechanical Methods of Dust Collection," by M. A. Lissman, *Chemical and Metallurgical Engineering*, vol. 37, 1930, pp. 630–634.

41 U. S. Patent No. 1,939,710; M. A. Lissman.

42 U. S. Patent No. 1,888,603; J. L. Mauthe.

43 U. S. Patent No. 1,915,436; T. V. Moore and H. D. Wilde.

44 U. S. Patent No. 1,773,840; A. Nattcher and F. Seipp.

45 U. S. Patent No. 1,826,086; R. G. Neumann.

46 U. S. Patent No. 1,928,702; R. F. O'Mara.

47 "Trennung von Staubgasgemischen durch Fliehkraft," by F. Prockat, *Glasers Annalen*, vol. 107, 1930, pp. 47–54.

48 U. S. Patent No. 1,795,888; H. Rabezzana.

49 "Modern Blast Cleaning and Ventilation," by C. A. Reams, Penton Publishing Company, Cleveland, O., 1939, p. 106.

50 U. S. Patent No. 2,015,464; E. C. Saint-Jacques.

51 U. S. Patent No. 1,994,049; E. C. Saint-Jacques.

52 "Verbesserung des Wirkungsgrades von Staubabscheidern (Zyklonen) in Entstaubungsanlagen," by A. Schaak, *Rauch und Staub*, vol. 22, 1932, pp. 127–128.

53 U. S. Patent No. 1,846,365; F. Seipp.

54 "Flow Pattern and Pressure Drop in Cyclone Dust Collectors," by C. B. Shepherd and C. E. Lapple, *Industrial and Engineering Chemistry*, vol. 31, 1939, pp. 972–984.

"Flow Pattern and Pressure Drop in Cyclone Dust Collectors; Cyclone Without Inlet Vane," by C. B. Shepherd and C. E. Lapple; *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 1246–1248.

55 U. S. Patent No. 1,920,437; D. A. Sillers.

56 U. S. Patent No. 1,509,910; A. H. Stebbins.

57 U. S. Patent No. 1,509,911; A. H. Stebbins.

58 U. S. Patent No. 1,845,566; A. H. Stebbins.

59 U. S. Patent No. 1,845,567; A. H. Stebbins.

60 U. S. Patent No. 1,950,069; C. M. Stein.

61 U. S. Patent No. 1,548,066; A. Steinbert.

62 U. S. Patent No. 1,906,432; C. E. Summers.

63 U. S. Patent No. 1,835,445; T. Thomson and N. Nisbet.

64 U. S. Patent No. 1,595,268; O. W. Van Petten and R. Deming.

65 "A Modern Dust Collector," by H. Van Tongeren, *Mechanical Engineering*, vol. 57, 1935, pp. 753–759.

66 U. S. Patent No. 1,940,196; F. H. Wagner.

67 U. S. Patent No. 2,005,305; F. H. Wagner.

68 U. S. Patent No. 1,591,147; S. S. Wales.

69 U. S. Patent No. 1,797,812; H. Waring.

70 U. S. Patent No. 1,825,377; G. W. Watts.

71 "Evaporation," by A. L. Webre and C. S. Robinson, Chemical Catalog Company, New York, N. Y., 1926, p. 108.

72 "Evaporation," by A. L. Webre and C. S. Robinson, Chemical Catalog Company, New York, N. Y., 1926, p. 467.

73 "Bemerkenswerte Rauchgasentstaubungen der Bauart Van Tongeren," by F. Wellman, *Feuerungstechnik*, vol. 25, 1937, pp. 108–116.

"Die Strömungsvorgänge in Zyklonen," by F. Wellman, *Feuerungstechnik*, vol. 26, 1938, pp. 137–145.

74 U. S. Patent No. 1,734,507; L. L. Westling and M. R. Hickman.

75 U. S. Patent No. 1,760,617; J. Whitmore.

76 "Performance Characteristics of Cyclone Dust Collectors," by L. C. Whiton, Jr., *Chemical and Metallurgical Engineering*, vol. 39, 1932, pp. 150–152.

"Results of Research on Cyclone Dust Collectors," by L. C. Whiton, Jr., *Power*, vol. 75, 1932, pp. 344–346.

77 U. S. Patent No. 1,798,510; C. A. Winslow and E. J. Hall.

78 "Some Factors and Principles Involved in the Separation and Collection of Dust, Mist, and Fume From Gases," by E. Anderson, Trans. American Institute of Chemical Engineers, vol. 16, part 1, 1924, pp. 69–85.

79 "Trennung von Staubgasgemischen durch Fliehkraft," by F. Prockat, *Glasers Annalen*, vol. 107, 1930, pp. 43–45.

80 "Grundlagen und Grenzen der Zyklonentstaubung," by P. Rosin, E. Rammler, and W. Intelmann, *Zeit. V.D.I.*, vol. 76, 1932, pp. 433–437.

81 "Note sur les dépoussiéreurs centrifuges," by M. Seillan, *Chaleur et industrie*, vol. 10, 1929, pp. 233–238.

Discussion

H. C. Moore, Jr.[4] In discussing the flow pattern in a cyclone, the authors state that they were unable to find the evidence of a secondary component of flow. They cite Shepherd and Lapple (54) as also being unable to detect the presence of the double-eddy current. It would be interesting to know what means were used to investigate whether or not this phenomenon was evident. Obviously, a visual observation alone is not always sufficient. The secondary component of velocity is usually of such magnitude that it affects only the finer particles of dust and/or liquid. These particles are of a size individually invisible to the naked eye and, when dispersed throughout a gas, just give a homogeneous appearance of darker color. With the velocities encountered in cyclones, it is quite impossible to determine flow direction, unless the individual particles are visible. The solution would be to use an extremely light dust wherein particles of corresponding terminal velocities have a larger dimensional size.

The existence of the double-eddy current in a fluid caused to flow in a curved path was observed by such early investigators as Boussinesq[5] and Reynolds[6] in the study of the flow of water. Lorenz[7] shows by mathematical analysis that, for a circular fluid flow, there is a difference in pressure between fluid streamlines at different radii. This pressure difference, together with the static-pressure gradient built up in the fluid due to the slower moving strata in the region of the enveloping walls, causes a displacement of streamlines into a pattern of two similar spiral flows known as the double-eddy current. It has been demonstrated that this double-eddy current is present in the flow of gases as well as liquids.[8]

In the design of cyclones, therefore, either of two alternatives is possible, i.e., the cyclone may be designed to prevent the formation of the double-eddy flow, or the inherent formation of the double eddy may be utilized to enhance the efficiency of collection. In the former case, the prevention of the natural formation is translated into increased turbulence, with a resultant impairment of efficiency and increase in pressure loss. The latter method is the more logical. In allowing the double eddy to take its natural form, the turbulence is thus reduced to a minimum, resulting in a minimum re-entrainment. However, if the double eddy is allowed to form, it is necessary to make some provision in the design to remove the dust which will concentrate at the top of the cyclone, and which, in coming down, passes in close proximity to the gas-outlet pipe. This is accomplished in the van Tongeren design by means of a shave-off slot at the top, which draws off this upwardly moving dust and passes it through an exterior duct for reintroduction into the cyclone proper at a point where the influence of the lower eddy is definitely downward.

In considering cyclones for separation of liquids, the effect of turbulence is probably not as great as in the separation of dust. Once the droplets reach the enveloping surface and become part of the film thereon, there is slight possibility of re-entrainment with proper design. However, in dust separation, a particle which has reached the wall of the cyclone is always subject to the possibility of re-entrainment due to turbulent flow.

[4] Moore-Broach Engineering Company, Atlanta, Ga.

[5] "Essai sur la theorie des eaux courantes," by J. Boussinesq, Mémoires Présenté par Divers Savants, vol. 23, Académie des Science, Paris, 1877, p. 596.

[6] "On Certain Laws Relating to Rivers," by O. Reynolds, British Association for the Advancement of Science, Report, 1887, pp. 555–562.

[7] "Der Widerstand von Rohrkruemmern," by H. Lorenz, *Physikalische Zeitschrift*, vol. 30, 1929, pp. 228–230.

[8] "Vortrage aus dem Gebiete der Aerodynamik und Verwandte Gebiete," by A. Gilles, L. Hopf, and T. von Kármán, Julius Springer, Berlin, 1930, p. 10.

A. L. Webre.[9] The results of this research are particularly interesting to the writer, inasmuch as he has used the so-called Webre cyclone for about 30 years in connection with the prevention of entrainment losses in pans and evaporators in the sugar industry, as well as in others—always with success.

In all of our standardizations, we have always used the vapor velocities in feet per second, instead of pounds per minute per square foot, used in the paper. Presumably, the experiments were conducted at atmospheric pressure. It is, therefore, pertinent to consider what would happen to the cyclone if the same weight of vapor were passed through at the higher lineal velocities encountered at high vacua. In line with this thought, whereas we have never recorded any research work done on the cyclones, yet we have had to standardize on certain proportions, and these standards have always been based on the actual linear velocities at the vapor inlet.

Thus we have found that we must not exceed certain limits. At 4 in. abs, we use 250 fps; if we exceed 275 fps, we begin to have entrainment losses. At lower vacua, and at atmospheric pressure, we reduce the velocity gradually, standardizing on 100 fps at atmospheric pressure.

If the velocity is too slow, then the apparatus begins to lose efficiency. In the case of black-liquor evaporators, where there is a great deal of foam, the whirling action of the cyclone is used to break up the foam. To accomplish this, the velocity must be in the neighborhood of 100 fps. It has very often been necessary for us to insert reducers at the cyclone inlet to accomplish this, without which, the foam would not be broken up.

In all cyclones, there is a loss of head between the inlet and the outlet due to the centrifugal force of the vapor itself, and this can be observed by placing manometers at the proper points. Under extreme velocity conditions, we have observed as much as 3 in. of mercury difference between the outer periphery of the cyclone and the center of the cover. It was mostly on this account that we have now abandoned the use of cyclones in favor of a different design which has no loss of pressure and performs with equal efficiency.

Particularly in the sugar industry, we have a very satisfactory means of determining the losses by entrainment, for all vapors are condensed. By using the alpha-naphthol test, we can determine as small an amount as 1 part of sucrose in 5,000,000 parts of water. With a well-operated sugar factory, the entrainment loss is less than the figure cited.

It might be stated further that the amount of liquid usually present in the vapor leaving an evaporator, and entering the cyclone, is relatively small. In the ordinary sugar evaporator, it would certainly be less than 2 per cent. In certain types of black-liquor evaporators, it is assuredly many times more than this figure, and a guess would be from 10 to 12 per cent by weight. The percentage by volume is naturally very small.

We should like to thank the authors for their illuminating work, and hope that they may be able to supplement this along the lines suggested, namely, under high vacuum and with smaller ratios between liquor and vapor.

[9] Mechanical Engineer, Merion, Pa. Mem. A.S.M.E.

Analyzing Governor-System Performance

By A. F. SCHWENDNER,[1] PHILADELPHIA, PA.

All the characteristics of a turbine governor considered as an entity have their effect on the operating performance, expressed in terms of speed-regulation sensitivity, rate of response, and stability. The governor can only be judged adequately by observing its operation in conjunction with the system with which it is associated, which can only be done by employing instruments more sensitive and rapid in response than the governor itself. This paper describes the essential features of an instrument, developed specifically for this purpose, the methods of testing employed, and actual tests made with it.

The turbine-generator-governing problem has attracted considerable attention in the last few years. Papers have been presented describing the effects of governor characteristics on the unit, station, tie line, and system. The desirable characteristics of governors have been pointed out, and the qualities of stability, sensitivity, and rate of response have been discussed.

Turbine users are mainly interested in the effect of governor performance on their systems. In other words, they are concerned with the operating-performance characteristics of their governors, for example, the unit-load changes corresponding to certain system-frequency changes as dependent upon unit loading. All the characteristics associated with a governor considered as an entity have their effect on the operating performance, i.e., speed-regulation (over-all and incremental) sensitivity, rate of response, and stability. In addition, the characteristics of associated units and the system-load conditions also affect operating performance. A governor cannot be adequately judged by observing only its behavior as an isolated mechanism or even by comparing the steam charts of different units. It must be carefully studied in conjunction with the system with which it is associated. Such study must be implemented with instruments more sensitive and rapid in response than the governor itself. An ideal set of such instruments would comprise a governor-performance analyzer capable of testing all the required governor characteristics in a short series of tests.

Governing-System Performance Characteristics

To discuss this subject intelligently there must be a well-defined nomenclature. The A.S.M.E. Power Test Code on speed-responsive governors does not cover all the terms and definitions used in recent years. The reorganized committee has not yet issued a new nomenclature. In the meantime something has to be used, but the terminology which has been adopted in this paper is not intended to establish new terms and definitions.

The operator judges the governor on the basis of its behavior when the unit is on the generating system. This behavior reveals what, for want of a better name, may be called the operating-performance characteristics of the governor. These are nothing more than the changes in load of the unit, corresponding to various classes of system-frequency change. The small frequency changes occurring in all systems, amounting to from 0.1 to 0.2 cycle, actuate the governing valves and change load to an extent depending upon the incremental speed regulation and sensitivity of the governor, for frequency changes slow in comparison to the rate of response of the governor. The operating performance approaches the incremental speed regulation as the sensitivity is increased.

A less sensitive governor or one with a rate of response slow, compared to the system frequency fluctuations or both will show a lower governor operating performance (effectively higher regulation) than the incremental speed regulation. Thus the operating performance of a governor can be measured by its deviation from the incremental speed regulation if the roughness or rate and magnitude of the frequency changes are also considered. The final decision on definition of and numerical expression of this deviation must be left to the joint A.S.M.E.-A.I.E.E. committee. The governor operating performance is so much dependent upon the individual governor-performance characteristics that all of them must be checked to enable us to analyze the former. Thus the following must be considered: Speed-regulation (over-all and incremental) sensitivity, rate of response, and stability.

Governor-Speed Regulation (Over-All). This is the sustained percentage of speed change of a prime mover for rated load change. Owing to the peculiarities of governors, this is not necessarily a unique value for a particular governor. The regulation for a no-load setting of the load-changing device may vary appreciably from the regulation for the full-load setting. Curves defining these variations are necessary to express fully the over-all speed regulation.

Governor-Speed Regulation (Incremental). This is the slope of the speed-load curve passing through 100 per cent speed at the load point in question, stated in percentage of rated load percentage of rated speed. If the governing valves had a linear steam-flow valve-lift characteristic and the turbine water rate were constant, the incremental and over-all regulations would be identical. The requirements for low valve pressure drops to obtain high efficiencies result in greater nonlinear flow-lift relationships.

Sensitivity. This is the speed change necessary to start movement of the governor and governing valves. It can be expressed in percentage of rated speed. It is important to be able to check the individual sensitivities of each of the elements of the governing system from the governor to the operating servomotor so that the source of insensitivity may be located.

Rate of Response. Perfect governing performance would be the exact following of speed by the governing valves. The governing valves actually lag behind their true speed position by an amount depending upon the shaft acceleration and the rate of governor response. The latter might, then, be measured in terms of the time required to initiate a corrective motion of the valves when the prime mover is accelerated at some constant and standard rate. Such an acceleration might be obtained by dumping an appropriate load. The effectiveness of a governor as applied to a particular unit is dependent not only upon its rate of response but also upon the inertia, rating, and speed regulation of the unit. This over-all effectiveness might be measured in terms of the time required to initiate corrective valve motion following the dumping of a definite fraction of the rated load.

---

[1] Control Engineer, Westinghouse Electric & Manufacturing Company, South Philadelphia Works.

Contributed by the Power Division and presented at the Semi-Annual Meeting, Kansas City, Mo., June 16–19, 1941, of The American Society of Mechanical Engineers.

It may be noted that broad transient regulation, introduced in some governors to obtain stability, will have a decided effect on the rate of response.

Stability. A stable governor is one which tends to damp rather than excite load or frequency oscillations. Small oscillations which may be revealed by a fast and sensitive recorder are not necessarily indications of fundamental instability.

Foregoing are the governor characteristics which should be measured to obtain a better conception of governor performance.

Specifications for a Governor-Testing Instrument

One of the first characteristics which should be demanded of a testing instrument is great flexibility. The many types of governor require an instrument readily adapted to the measurements of their various characteristics. This adaptability should be obtained without major alterations of the parts of the instrument. By using unit sections for the measurement of the different governor functions, the different elements can be interchanged quickly. It is also important to have a record of all the elements on a single chart. It is essential to be able to measure the following quantities: (*a*) Angular velocity of the turbine rotor; (*b*) travel of the governor and servomotors; (*c*) hydraulic pressures corresponding to motions in flyball governors; and (*d*) electrical output of the generator. A governor-testing equipment should be suitable not only for an initial determination of the governor-performance characteristics, but also for periodic checks to determine when the slow depreciation in performance justifies taking the governor out of service for repairs.

Equipment which may be used frequently should be designed with light compact units arranged so that the complete setup would require only a few hours. In a large power plant it should be possible either to move the instrument as a unit from turbine to turbine or to locate it centrally and provide for cable connections to the measuring elements which may be located on the various machines. The elaborate precautions necessary in transmitting and magnifying motions mechanically with levers and wires, and the difficulties in moving such an arrangement from unit to unit, suggest the advantage of transforming the motions into proportionate electrical quantities directly at the source.

Similarly, pressures are conveniently converted to electrical units and transmitted by electrical cables which are easily handled and laid. The pressure and travel elements should be so designed that after the type of elements has been selected as dictated by the governor to be tested, all the tests can be completed without changing the elements. Each element should have sufficient range so that the changes in characteristics, required for the different tests, can be obtained with easy and rapid adjustment. If the adjustments are obtained in definite steps, recalibration after a change may be omitted. The individual measuring elements should indicate the smallest significant change of the measured quantity without overtravel or time lag, regardless of the instrument magnification ratio. Analysis of records is considerably facilitated if the orders of magnitude of the deviations of the measured quantities as recorded are roughly equal, whatever their actual values.

The recording instrument can be a standard oscillograph. Direct recording with pen or stylus on a paper chart would be preferable if a multielement recorder with sufficiently high frequency response were available. Development of such an instrument has been interrupted by more pressing work. For a typical governor test, one frequency element, three travel, a load, and a timing element for a mechanical governor, or one frequency, two pressure, one travel, a load, and a timing element for an oil governor are required as a minimum. To analyze practically any governing system, one frequency element, three pressure, three travel, several load elements, and a time element would cover all requirements. A long-film seven-element oscillograph would be adequate for most purposes. A range of film speeds from $^1/_8$ to $^1/_2$ in. per sec to 3 in. per sec and an easily adjustable light intensity are desirable.

Speed-Measuring Element

The individual measuring elements should have the following characteristics:

The speed-measuring element should be arranged for attachment to the shaft end wherever possible. In many cases, the shaft ends are not available for this purpose. For this reason a synchronous motor, connected to the unit through the instrument transformers, can drive the speed-measuring element at an angular velocity reasonably close to that of the turbine. Error due to variations in phase angle of the terminal voltages and to electrical flexibility between generator and motor will limit this method to those cases in which such error is permissible.

Two full-scale speed ranges are required: A wide range from approximately 50 to 70 cycles per sec and a sensitive scale with full-scale deflection for about 0.25 cycle from a base frequency which may be set anywhere from 59 to 61 cycles. The full-scale deflection should be sufficient to show definitely a speed change of 0.01 per cent or less on the sensitive scale.

Motion-Measuring Element

To follow and record the motion of any part of the governing system the measuring element should have a substantially flat frequency response to about 60 cycles per sec. In order to obtain the required response, it is desirable to have a short, stiff connecting wire between the moving part and measuring element. Converting the motion into proportional electrical current permits mounting the element very close to the moving part. The element should be capable of a wide range of ratio adjustment to be able to record the different motions in a governing system. Besides the wide-range ratio required to give full-scale deflection for the total stroke of a movement, a sensitive range is also needed to give full-scale deflection to a motion corresponding to a change in frequency of 0.25 cycle. The changeover from one ratio to the other should not require more than a few minutes and the element should be so constructed that it can be attached to any part of a governing system without trouble. Some protection should be provided to allow sudden overtravels of the moving part, while the element is set to the sensitive range without damaging the element.

Pressure-Measuring Element

The oil pressures to be recorded for the purpose of analyzing governor performance vary about as much as the motion of the governing system. The pressure changes and ranges require a wide ratio of adjustment. Most of the pressures also require two complete ranges, a wide range which will record the deflection corresponding to the maximum pressure change, and a sensitive scale which will give full-scale deflection to an oil-pressure change corresponding to a change in frequency of 0.25 cycle. It is possible to develop an adjustable-ratio pressure-measuring element, or two elements may be used, having the proper ratios to permit analysis of the governing-system performance, using the same gage line with suitable cutover valves. In any case, the arrangement should be such as to permit changeover from one ratio recording to the other without appreciable loss of time. Wide variation in base pressures is also required.

Load-Measuring Element

The polyphase average-watt oscillograph galvanometer which is available for use in standard oscillographs measures true watts

in three-phase circuits. Such a galvanometer may advantageously be employed as the load-measuring element. The accuracy and response of this element must be matched reasonably closely to those of the other measuring elements. It is quite a task to draw up specifications for a universal governor-performance testing equipment. The specifications outlined cover the elements required to evaluate the governor-operating performance characteristics and to make all the other tests required to establish those characteristics. The elements described can be used to investigate and analyze governing and generating system performance, or tie-line characteristics, by using the proper combination of these elements. It is advisable to have all the elements so made as to be able to record both direct and alternating components with frequency-response characteristics substantially flat from zero to 50 or 60 cycles per sec (if feasible).

Testing a Governing System

Given an instrument which can readily be adapted to any unit for a quick determination of governor characteristics, we must also have a test program which will enable us to obtain all the necessary information with the least possible outage or system disturbance. The test method to be outlined is simple to follow and adequate to provide all necessary information to make possible an analysis of governing-system performance:

1 With the unit on the line and carrying maximum load, measure the speed-changer and governing-valve positions. Then reduce the load in small increments, preferably with a load-limiting device. Hold every load position long enough to run a short record on the oscillograph or to read the deflections directly. The steam conditions must, of course, be nearly identical for all of these readings. When rated load is reached, record also the speed-changer and governing-valve positions necessary to sustain this load at rated speed. Remove the remainder of the load in the same manner.

2 Take the generator off the line and, with the recorder running, move the speed changer from the no-load to rated- and maximum-load positions. Then return it to the no-load, full-speed position. Partly close the throttle valve and so drop speed until the governing valves open to the rated-load position, and then to the maximum-load position. Reopen the throttle valve and bring the turbine speed back to normal. While the speed changer is being moved, the steam pressure and vacuum should be kept constant within reasonable limits.

3 Synchronize and put the generator back on the line with the recorder running.

4 While the load on the unit is being brought up to its previous value, run the recorder for a short time at all settled load points, preferably identical with load points of the first test.

The first test requires only the load and governing-valve-travel measuring elements of the analyzing instrument. Both are set to the wide range and the recorder is run at a slow speed. The second test requires the speed and governing-valve-travel measuring elements. The governor-travel or governor-oil-pressure measuring element can be added to the others. All the elements are to be set to the wide recording range with the recorder set to run at a slow speed.

The third test requires the speed, governor-travel or pressure, and governing-valve-travel measuring elements. The load element can be added for a better over-all picture. All the elements are to be set to the sensitive recording range. The recorder should be set to run at high speed.

The fourth test requires all the governor-travel- or pressure-measuring elements with the speed-and-load element included.

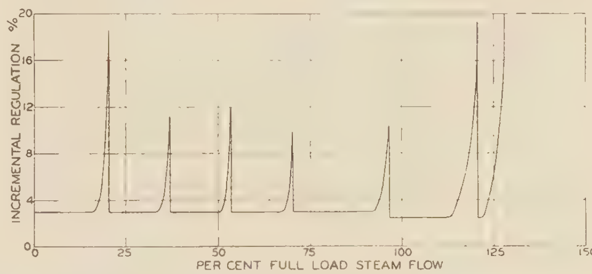


Fig. 1 Steady-State Incremental Regulation Vs. Steam Flow

All the elements are to be set to the sensitive recording range. The recorder should be set to run at slow speed.

Having the specification for the instrument and also having determined the program for a governor test, we now need the instrument itself and some test results to prove that the type of instrument and the method of test will provide all the required information about governor performance.

Some time ago J. E. Allen (1)[2] built an instrument which very

[2] Numbers in parentheses refer to the Bibliography.

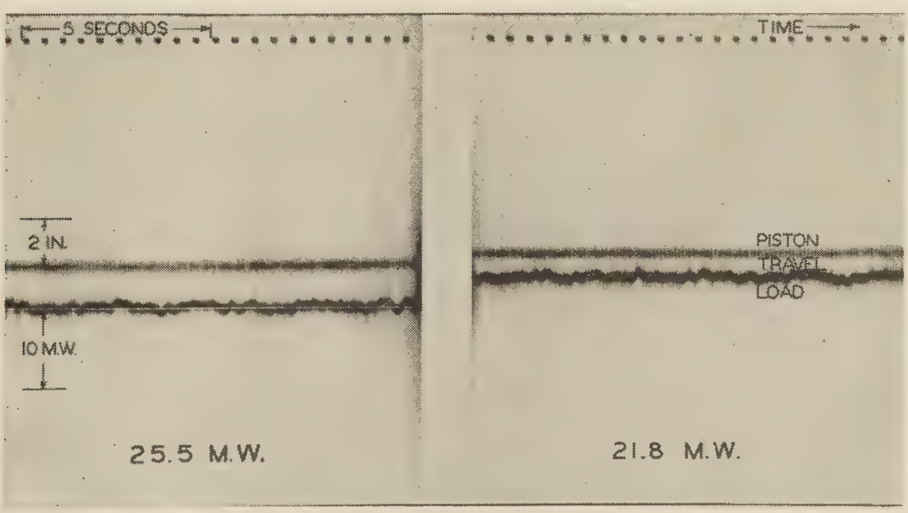


Fig. 2 Typical Records of Load and Valve Travel

nearly fulfills our requirements. The East Pittsburgh research laboratories have since built a governor-analyzing instrument which satisfies the specifications laid down. It will not be necessary to enter into a detailed description of this instrument, as it will be covered in a paper by W. O. Osbon (2).

Analyzing Test Results

We set out to check what the governing system of a specific unit will do when connected to a power-generating system. The results obtained should be prepared in such form that the operators will know the incremental regulation at each load point, for each unit in their system. For good system regulation it is important to know at any time at what regulation each component of the total system capacity is being generated. For that purpose a curve showing the load points of the unit plotted against the regulation at each load point will be the most useful. Fig. 1 shows a typical steam-flow incremental regulation curve. When the test results obtained with the analyzing instrument are worked up, we shall obtain a similar curve except that it will show load incremental regulation. It will, of course, also differ by the amount of the deviations due to change in water rate. The information obtained by the test can be used to plot the incremental regulation curve. Two methods are available and they can be used to cross check the results obtained.

The records of a test run taken under the conditions described in test paragraph 1 are shown in Fig. 2. The load and valve or governor travel obtained in this manner can then be plotted in curve form, as shown in Fig. 3. The curve shows steam flow, while the test will give load and the corresponding valve or governor position.

The curve shown in Fig. 3 can be transformed into the shape shown in Fig. 1 if the over-all speed regulation at the particular load point is known. The over-all speed regulation was checked during the second test.

The test results are best plotted in a curve of the shape shown in Fig. 4. The speed-changer travel is plotted against the percentage of normal speed. The speed increase, recorded while the

Fig. 3 Governor-Travel Versus Steam-Flow Characteristic of Multivalve Steam Flow

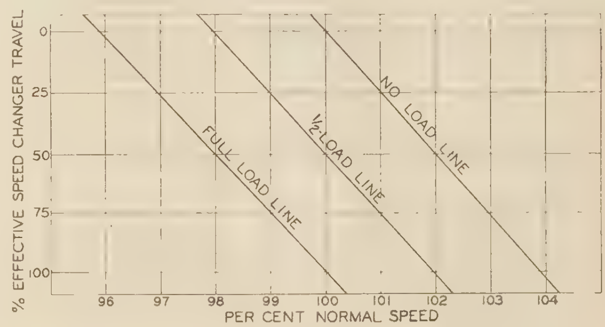


Fig. 4 Governor-Regulation Characteristic

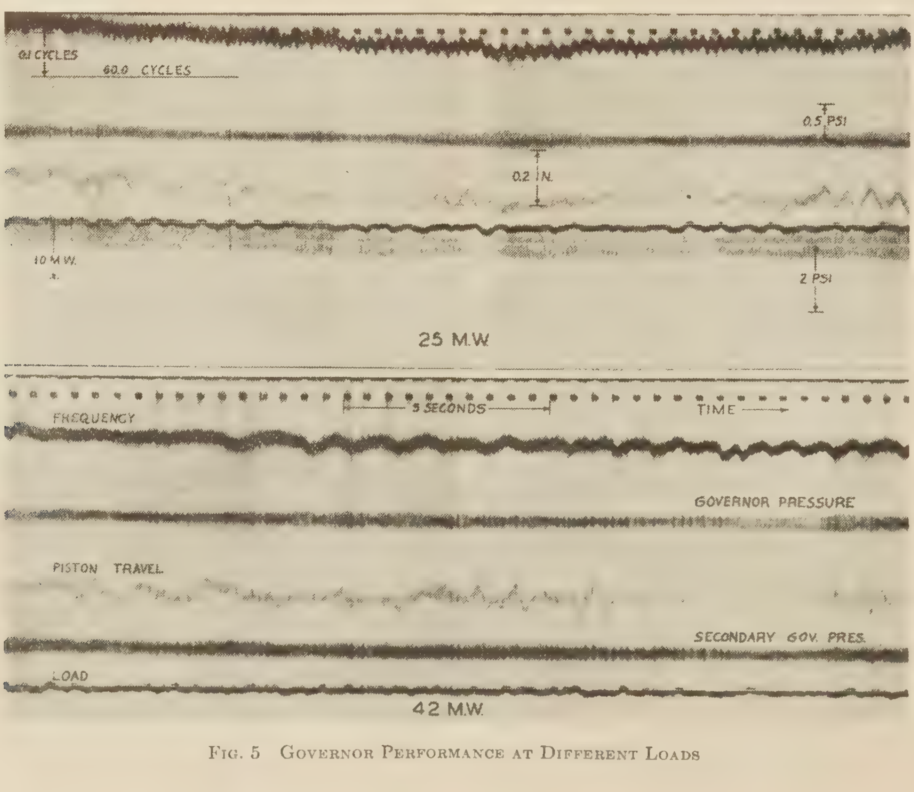


Fig. 5 Governor Performance at Different Loads

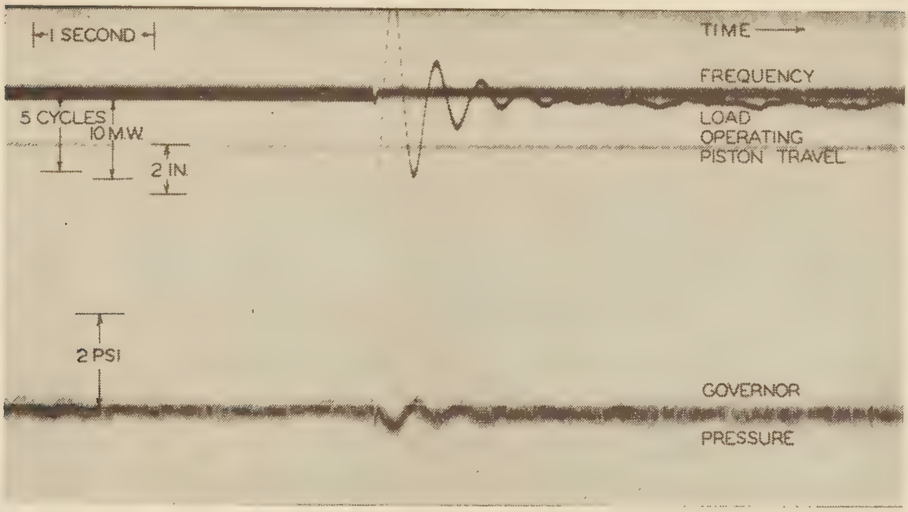


Fig. 6 Synchronizing Transients

speed changer was moved from the no-load to the maximum-load position, should be marked on the chart and thus will become the no-load line.

The drop in speed which occurred when throttling until the maximum-load governing-valve positions were reached should be plotted on the horizontal zero line of the "effective speed-changer-travel" axis. Connecting this point with the 100 per cent speed-travel point gives us the full-load line and completes the chart. Fig. 4 shows almost parallel no-load full-load lines. Most of the travel governors have considerable variation between the two points. The actual over-all regulation for any load point should be taken from the horizontal line between the no-load full-load lines at the point where the particular load line crosses the 100 per cent normal speed line.

The second method of obtaining governor-operating characteristics is by analysis of the records obtained by the test outlined in test paragraph 4. Two records obtained at two load points are shown in Fig. 5. Even a casual observation of the two records shows that, while the frequency changes have been followed by the governing system, the load change has not followed them. The maximum frequency change of 0.07 cycle is followed by a valve travel of 0.11 in. and a load change of 1500 kw in the top record. The lower record shows a maximum frequency change of 0.05 cycle with a valve travel of 0.08 in. and practically no change in load.

If the changes in frequency are plotted against changes in load and a line drawn through to average all the points, the slope of the line will give us the incremental regulation for that particular load point. A considerable number of additional cases will have to be analyzed before it will be possible to evaluate the width of the band obtained.

J. E. Allen has developed a governor-analyzing instrument which records the frequency on one axis and the valve movement on the other axis simultaneously. This type of record would eliminate the rather cumbersome plotting of points, but it neglects the effect of valve-flow characteristics.

The stability of a governing system can be shown by means of the records obtained in the test indicated in test paragraph 3, as shown in Fig. 6. Frequency and governor pressure show no tendency toward oscillation up to the point of synchronization.

When checking sensitivity down to very small values, it is difficult to obtain definite results. Any ripple in a recorded quantity, regardless of its cause, reduces the accuracy and definiteness of the result. The frequency and motion records, shown in Fig. 5, indicate a motion with a frequency change of from 0.01 to 0.02 cycle.

To establish the rate of response of a governing system, it is necessary to arrange for a load-dump test. The instrument described is also suitable to obtain the type of records necessary to establish the rate of response in a load-dump test.

Conclusion

The test results so far obtained demonstrate that the instrument, built in accordance with the specifications given in this paper, is capable of analyzing the governor performance of almost any type of governor. The instrument can be set up to test a governor in about 3 hours, and the test itself requires less than 4 hours. Most of the tests thus far made were taken only recently, and not all of the records have been evaluated. Where test values were not available, calculated values have been substituted.

The sensitivity and rate of response of the few governors tested, in comparison with the rate and magnitude of the frequency changes, did not give us a measurable deviation from the incremental speed regulation at the few check points made. Further tests are required on units which will show a measurable deviation between the incremental speed regulation and the operating performance of the governor. Only after the completion of such a test can the method of testing and evaluating test results be established and completed information presented.

Acknowledgment

The author wishes to express appreciation for the assistance given by W. A. Wilson of the Steam Division of the Westinghouse Electric & Manufacturing Company in preparing this paper.

BIBLIOGRAPHY

1 "Oscillograph Analyzes Governor Performance," by J. E. Allen, *Power*, vol. 78, 1934, pp. 610–612.

2 "A Turbine Governor Performance Analyzer," by W. O. Osbon, Transactions A.I.E.E., vol. 60, 1941, November Section, pp. 963–967.

3 "Regulation of System Load and Frequency," by H. Estrada and H. A. Dryar, Trans. A.S.M.E., vol. 62, 1940, pp. 221–232.

4 "The Field of System Governing," by A. Davis, Trans. A.S.M.E., vol. 62, 1940, pp. 207–217.

5 "Prime-Mover Speed Governors for Interconnected Systems," by R. J. Caughey and J. B. McClure, *Electrical Engineering*, April, 1941, Transactions section, pp. 147–150.

Silica Removal by an Improved Magnesia Process

By H. L. TIGER,[1] NEW YORK, N. Y.

Modern high-pressure boilers with high feedwater make-up requirements frequently demand reduction of dissolved silica. A new magnesia process for silica removal has been developed, combining flexibility, simplicity, and economy. It can be applied to any type of water at any temperature; it can be carried out in conjunction with the usual precipitation water-softening process; and the increase in operating cost is very small.

These results are accomplished by a new method of combining a number of principles embodied in this process. Extensive investigations were conducted to determine the influence of each of various factors on the final result, and data are presented on the effects of the following: Form of magnesium used (whether precipitated ionic Mg or undissolved Mg compound), temperature, sludge-magnesium concentration, and agitation for sludge-water contact. The principles of increasing the Mg^{++} content, when required, by means of a novel sludge recycling process with a magnesium dissolver are described. A method of using cheap dolomitic lime as a low-cost source of magnesium is also presented.

The process lends itself advantageously to a modified Spaulding precipitator type of construction at either low or high temperature, to be followed by suitable treatment with carbonaceous-ion exchangers, when required, for complete removal of residual hardness and reduction of alkalinity to any predetermined figure.

THE trend toward higher pressures and higher percentages of make-up in steam boilers and toward lower alkalinites in boiler feedwater has resulted in siliceous deposits in boilers and turbines in several cases. Boiler-tube failures, due to such deposits, are described in many papers (1, 2, 3, 4, 5, 6).[2] Troubles from these deposits in superheaters and turbines, as a result of carry-over of silica-containing salines with the steam, are described in other papers (7, 8, 9, 10, 11, 12, 13). This has caused a demand for an effective and economical method of reducing silica in boiler feedwater or otherwise preventing silica deposits.

Dissolved silica is always present to some extent in all natural waters. Suspended silica is also present in surface waters containing turbidity. Such suspended silica or siliceous substances can be readily removed by the usual coagulation and filtration methods. It is the silica remaining in solution which offers the special problem with which we are concerned in this paper.

The factors which influence the formation of silica deposits in boilers and the nature of the deposit which forms have not been thoroughly explored. It is known, however, that the presence of calcium and magnesium in the feedwater tends to form calcium and magnesium silicates. The presence of aluminum tends to form alumino-silicates, of which analcite is a special form. Under certain conditions, pure silica alone may constitute the bulk of the deposit.

The "silica tolerance," i.e., the maximum allowable amount of silica that may be tolerated in the concentrated boiler saline, cannot be specified with accuracy at present, due to the lack of sufficient statistical information on that subject over a sufficiently broad range of boiler pressures, ratings, etc. This is a subject which deserves further investigation by our engineering societies. If the feedwater contains little or no dissolved calcium, if the ratio of alkalinity to silica is sufficiently high, for example, more than 1:1, and if the boiler circulation is satisfactory, the silica content of the boiler salines may be as high as 350 ppm. Several cases are on record where such conditions prevail in high-pressure boilers without resultant difficulties. In some cases when silica-scale troubles started, they have been overcome by increasing slightly the ratio of alkalinity to silica in the boiler saline. In other instances, such difficulties have been overcome by improving boiler circulation to avoid "hot spots" or "dry tubes," i.e., local areas in which the steam is formed more rapidly than it is carried away by the circulating water in the boiler.

The trend in modern boiler-feedwater-conditioning practice is to reduce the hardness and total solids to the lowest possible limits and also to reduce the alkalinity to a minimum in order to avoid carry-over and CO_2 in the steam. Such modern methods of feedwater conditioning have been described in previous papers (14, 15, 16). These methods are applicable even where raw water contains sodium bicarbonate which hitherto could be removed only by distillation (17).

This trend toward lower alkalinity has created a demand for a corresponding reduction in the silica content, because if the "silica tolerance" limits the amount of silica in the saline, then the amount of boiler blowoff is controlled in many cases by the silica in the feedwater rather than by the other solids present. The silica in the feedwater is concentrated along with other substances which are present in solution. The extent of such concentration can, of course, be controlled by regulating the amount of boiler blowoff which is replaced by a corresponding amount of fresh feedwater. However, if the silica in the feedwater is very high and the "silica tolerance" in the saline is very low, such control of the silica in the saline by blowoff becomes difficult and expensive. These considerations have led to a demand for an efficient method of reducing silica. However, it must be realized that the cost of reducing silica increases disproportionately with each additional ppm reduction as the silica desired in the effluent approaches zero. Therefore, a balance should be determined between the boiler blowoff and silica reduction by treatment of the feedwater to arrive at the most economical and satisfactory method of keeping below the silica tolerance.

If, for example, the boiler saline must be limited to total solids 1500 ppm and silica 30 ppm, and the total solids in the make-up

[1] Vice-President in charge of Research and Development, The Permutit Company. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Joint Research Committee on Feedwater Studies and presented at the Annual Meeting, New York, N. Y., December 2-6, 1940, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

after treatment is 200 ppm, then the desired SiO_2 in the make-up water is calculated as follows

$$\frac{1500}{200} = 7.5 \text{ concentrations}$$

$$\frac{30}{7.5} = 4 \text{ ppm silica}$$

It is then only necessary to reduce the silica to 4 ppm in the make-up, since the amount of boiler blowoff required to maintain total solids below 1500 ppm will at the same time maintain silica in the boiler saline below 30 ppm.

Analytical Methods and Literature References to Silica Removal

The method used throughout the investigation described in this paper was that adopted by the A.P.H.A. (18) with some modifications. Swank and Mellon (19) buffered-chromate standards were used, because our tests confirmed the higher accuracy obtainable with these standards. This agrees with the findings of Knudson, Juday, and Meloche (20).

We also found it necessary to make all our silica determinations at 20 to 25 C, since variations in temperature affected the color of the silicomolybdate complex. This was especially noticeable in cold-water investigations, where determinations at the low temperature gave false low readings.

Experience showed also that glass vessels might seriously contaminate the solutions, particularly when they were highly alkaline and at high temperatures. To determine the extent of such contamination, tests were made with various types of containers, periods of contact, temperatures, and solution alkalinities. These tests led to the following conclusions:

(*a*) Hot-alkaline samples require suitable metal or rubber or other nonsiliceous containers.

(*b*) Pyrex glass is fairly satisfactory.

(*c*) Soft-glass containers may add considerable contamination and should only be used on cold waters of fairly high silica content and fairly low pH value (below pH = 8.3, the change point for phenolphthalein indicator).

M. C. Schwartz made an excellent study of the analytical methods for silica determination, and his paper (21) contains a comprehensive summary of the literature and a bibliography on this phase of the subject. Recently Kahler (22) presented data on improvements in the use of the Isaacs method which depends upon the blue color of the reduced silicomolybdate complex.

Very little was published on the removal of silica from water prior to 1930. Exploratory experiments were made before that time, but no practical processes were evolved (23, 24, 25).

Since 1930, a considerable amount of material has been published describing various processes. Most of these processes depend upon the removal of silica by absorption[3] or adsorption[3] by hydrous metallic oxides or hydrated oxides, such as, alumina, ferric oxide, magnesium oxide, zinc oxide, and other miscellaneous metal oxides.

In some of the processes described, the water is filtered through beds of metallic-oxide granules, which may or may not be capable of regeneration; in others, the metal oxides or hydroxides are precipitated in the water; while in others, the solid absorption materials are fed to the water. Processes have also been suggested in which calcium and magnesium aluminates are employed to precipitate the silica from the water.

[3] The more general term "absorption" is used throughout this paper, as it is possible that these processes involve other phenomena besides straight adsorption.

Silica removal by passage of water through beds of metallic-oxide granules has been described by Liebknecht (26), Kurz (27), but this method has not been adopted in practice because of high operating cost and operating difficulties.

Silica removal by precipitation of aluminum hydroxide at high pH and by various aluminates is described in detail by Christman, Holmes, and Thompson (28), Stumper (2, 29), Black, Bardwell, and Graham (30), Kaissling (9), Lindsay and Ryznar (31), Lindsay and Braithwaite (32), and Betz, Noll, and Maguire (33). However, high treatment cost, as well as the possibility of analcite-scale troubles in the boilers, due to residual alumina in the treated-water effluent, has discouraged the adoption of the aluminum-hydroxide and aluminate methods of silica removal.

Silica removal by precipitation of hydrous ferric oxide from ferric sulphate has been described by Smith (24), Stumper (2), Schwartz (34), Powell, Carpenter, Setter, and Coates (36), Behrman and Gustafson (37). The article by Schwartz (34) describes the characteristics of this hydrous-ferric-oxide process in detail. Several large boiler-feedwater-treatment plants, employing this process, have been in operation for a number of years with satisfactory results, Applebaum (15). However, the cost of treatment is high, especially when the initial silica in the water is high. Furthermore, the precipitation of ferric hydroxide from the ferric sulphate fed results in an increase of soluble sulphates, i.e., an increase in the total-dissolved-solids content of the boiler feedwater, which is frequently undesirable.

Reimers (38) suggested the precipitation of hydrous zinc oxide, cadmium oxide, manganese oxide, etc., but this process was never adopted in practice because of high operating cost.

Silica removal by magnesium oxide has been referred to frequently in the literature. Bohlig (39) referred to the use of calcined magnesite for reducing the calcium-bicarbonate content of water and noted that silica is also removed from the water in the same treatment. Stumper (29, 2) describes beaker tests on silica removal by calcined magnesite at high and low temperatures and concludes that MgO is of no value at room temperature but may be of some use at boiling-water temperatures. Christman, Holmes, and Thompson (28) present some laboratory test data indicating silica removal by precipitation of ionic magnesium at room temperature and at boiling-water temperatures. Hundeshagen (3) and Kurz (27) also recommended the removal of silica by precipitating $Mg(OH)_2$, or by the addition of calcined magnesite (MgO). Kuhnert (40) recommended the use of MgO and $MgCO_3$ for removing silica from natural brines. Straub (41) studied the removal of silica from boiler salines at 182 to 282 C by the addition of solid MgO inside the boiler. Geisler (7) fed dried precipitated MgO to the water to remove the silica in a hot-process softener and claimed excellent results. Wesly (42, 13) stated that phosphate ion interfered with the absorption of silica by solid MgO at high temperatures. Peters and Turner (43) recommend that partial lime softening of boiler feedwater be employed in order to avoid precipitating the Mg^{++} in the water softener. Then this Mg^{++} is precipitated in the boiler, and they state that this precipitation reduces the SiO_2 content of the boiler salines to very low values. Betz, Noll, and Maguire (44, 45) recommend the use of special forms of magnesium oxide for removal of silica in the hot-process treatment. Low-temperature treatment by magnesium oxide is not recommended by these investigators, even though it is claimed that longer retention periods will improve silica removal at low temperatures. Behrman and Gustafson (37) recommend the precipitation of $Mg(OH)_2$ for removing silica in cold treatment and suggest feeding magnesium carbonate or magnesium bicarbonate to increase the ionic Mg content of the water being treated without leading to an increase in total dissolved solids of the treated

water. They state that no improvement in silica removal is obtained by the use of dolomitic lime [$Ca(OH)_2 \cdot MgO$] instead of ordinary lime.

It is interesting to note that, despite the activity in this field during the last decade, no data have been presented hitherto on a widely applicable silica-removal process. On the other hand, the new magnesia-silica-removal process to be discussed presently has already been embodied in a large-scale installation, which has been operating for more than a year, and it may be added that further installations embodying this process are under construction at the present time.

Development of Magnesia Process for Silica Removal

The investigations discussed in this paper were carried out with a view to developing a silica-removal process that would

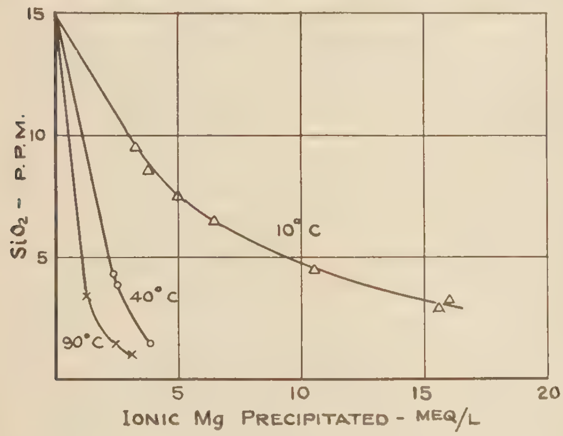


Fig. 1 Silica Removal by Ionic Mg Precipitated at Various Temperatures

Note: (a) Raw well water containing 15 ppm SiO_2; various amounts of Mg^{++} added in form of $Mg(HCO_3)_2$ or $MgCO_3$.
(b) Agitated with alkali for 30 min to precipitate the ionic Mg; decanted and filtered through paper, and SiO_2 determined on filtrate.
(c) Precipitated Mg^{++} only; no sludge added.

be simple and foolproof, in addition to being economical and sufficiently flexible to satisfy the wide variety of conditions which result from the various combinations of the following factors: Composition of the raw water, silica desired in the effluent, temperature of water to be treated, volume of water involved, other treating processes which must be applied to the water in addition to silica removal.

In order to determine the possibilities of developing such a process, several series of exploratory experiments were made during the early stages of this research. These investigations indicated that it would be possible to develop a simple and economical process and one that would be sufficiently flexible to meet the varying conditions enumerated. It was realized, however, that before such a process could be evolved, it would be necessary to carry out a systematic study of the influence of each of a number of factors affecting the process and to correlate these properly in designing equipment to suit each particular set of operating conditions.

The various aspects of the problem investigated include:

1 Silica removal by precipitating ionic Mg:

(a) Effect of temperature and agitation
(b) Methods of increasing the ionic Mg content of the raw water when that is necessary
(c) Most suitable and economical sources of Mg in such cases.

2 Silica removal by solid, undissolved Mg compounds:

(a) Effect of temperature and agitation
(b) Influence of sludge; type of sludge, concentration, and degree of contamination
(c) Most suitable and economical sources of Mg.

3 Silica removal by combining the effects of precipitation of ionic Mg and undissolved Mg compounds.

Precipitated Ionic Magnesium

Experiments were made to determine the controlling factors governing the silica reduction by precipitating ionic Mg, and in this series of tests, the sludge was eliminated in order to observe the effect of ionic Mg precipitation alone.

The Mg was added in the form of soluble $Mg(HCO_3)_2$ or $MgCO_3$ to a clear well water containing about 15 ppm SiO_2. The Mg was then precipitated by adding alkali and agitating for about 1 hr. These tests were also extended to various temperature ranges. The initial and final Mg and SiO_2 were measured, the results being summarized in Fig. 1.

These data show that:

(a) Silica reduction increases with increasing amounts of ionic Mg precipitated at any given temperature.

(b) The efficiency of silica absorption by precipitating ionic Mg increases greatly with rise in temperature.

It is apparent from these data that, from the technical point of view, it is feasible to reduce the silica to any desired extent by precipitating ionic Mg, but a number of questions immediately present themselves. What if there is not sufficient Mg in the raw water to permit precipitating the required amount of Mg for the desired silica reduction? In that event, how should it be added and in what form, and from what source should the Mg be obtained? Should the water be heated in order to take advantage of the great increase in silica-removal efficiency with rise in temperature?

The answers to these questions obviously involve considerations of design, availability and costs of Mg compounds, and local costs of heat. In order to study these matters adequately, it is first necessary to examine the reactions involved in the precipitation of ionic Mg and the dissolving of Mg compounds.

Principles of Dissolving and Precipitating Mg Compounds

The simplest method of adding Mg^{++} is to use a readily soluble salt, such as $MgSO_4$ which is usually available on the market in the form of Epsom salt, $MgSO_4 \cdot 7H_2O$. When Mg ion is precipitated from an $MgSO_4$ solution by a hydroxide like lime, the following reaction takes place

$$MgSO_4 + Ca(OH)_2 \rightarrow Mg(OH)_2 + CaSO_4 \ldots\ldots [1]$$

The insoluble $Mg(OH)_2$ is precipitated, and the $CaSO_4$ is left in solution and is subsequently converted to a corresponding amount of Na_2SO_4 by the other steps in the treatment which are applied for removing the residual hardness. Thus each milliequivalent (meq) of $MgSO_4$ added leaves 1 meq of Na_2SO_4 in solution. The molecular weight of Na_2SO_4 = 142, and its equivalent weight $= \frac{142}{2} = 71$. Therefore the addition of 1 meq of Mg per l in the form of $MgSO_4$ (weighing 60 mg) increases the total solids of the final effluent by 71 mg per l or 71 ppm.

If, on the other hand, the Mg^{++} is added in the form of $MgCO_3$ or $Mg(HCO_3)_2$ and is then precipitated by hydrate of lime, the reactions are as follows

$$MgCO_3 + Ca(OH)_2 \rightarrow Mg(OH)_2 + CaCO_3 \ldots\ldots [2]$$

$$Mg(HCO_3)_2 + 2Ca(OH)_2 \rightarrow Mg(OH)_2 + 2CaCO_3 + 2H_2O \quad [3]$$

Both the reaction products, $Mg(OH)_2$ and $CaCO_3$, are relatively insoluble, and thus the dissolving of the Mg in this form and its subsequent precipitation do not leave any salts in solution, i.e., do not increase the total dissolved solids. It should be borne in mind, however, that commercial magnesite (which is the only fairly cheap form of $MgCO_3$) cannot be dissolved readily in water. But calcined magnesite (MgO) can be dissolved readily if brought into a fairly low pH environment, such as that prevailing in the usual raw water where HCO_3^- and CO_2 are present. When MgO is brought into contact with such a water, the following reactions take place

$$MgO + CO_2 \rightarrow MgCO_3 \quad [4]$$

or

$$MgO + 2\,HCO_3^- \rightarrow MgCO_3 + CO_3^= + H_2O \quad [5]$$

The reactions are essentially the same for dissolving $Mg(OH)_2$ which, as shown in Equations [2] and [3], is the form in which Mg is precipitated from solution.

Equations [4] and [5] show that the amount of Mg which can be dissolved from MgO depends upon the $HCO_3^- + CO_2$ content of the raw water. Where the raw water does not contain enough of these constituents to dissolve the required amount of Mg, CO_2 may be added readily and economically to the raw water by means of flue gas, in order to increase the MgO dissolved per Equation [4]. It is necessary to add a constant amount of CO_2 per unit volume of water for dissolving a constant amount of magnesium per unit volume of water. With this in mind, two arrangements may conveniently be used:

(*a*) *Presaturation of the Raw Water With CO_2.* This is accomplished by using a trickler-carbonator or bubbler arrangement, in which an excess of flue gas is contacted with the water to saturate all or a definite portion of the water with a constant amount of CO_2 under any given set of conditions. The water containing this constant amount of CO_2 is then admitted to the magnesium dissolver, where it is agitated with an excess of undissolved MgO.

(*b*) *Simultaneous Carbonation and Mg Dissolving.* This is carried out by using a flue-gas bubbler with measuring pump (compressor) to discharge a measured volume of flue gas directly into the agitated suspension of MgO in the magnesium dissolver. This dissolver is so designed that all the CO_2 added is absorbed by the water in the magnesium dissolver. Method (*b*) is useful when it is not possible to introduce sufficient CO_2 in the presaturation carbonator of (*a*) to dissolve the required amount of ionic Mg in the magnesium dissolver.

Sources of Ionic Magnesium

There are available several Mg compounds which are soluble directly in water containing HCO_3^- and CO_2. As mentioned, some of these, like $MgSO_4$, dissolve readily without reacting with the HCO_3^- and CO_2, while others, like MgO, require HCO_3^- and CO_2 to carry them into solution. In addition to the disadvantage of increasing total solids, as previously discussed, $MgSO_4$ has a high cost per unit Mg, as shown in Table 1.

In addition to MgO, there is another source of Mg which is even much less costly, namely, dolomitic lime. This product is made by calcining dolomite, and its composition is shown in Table 1. Although hitherto it has been customary to specify lime-free from Mg for water treatment (46), it is now found that these high Mg limes are a convenient and economical source of MgO for silica removal. The figures in Table 1 show it to be the cheapest source of Mg, after duly crediting the cost of the dolomitic lime with the value of the lime contained therein. This applies where the lime present in the dolomitic lime can be usefully consumed in the softening or other reactions. When the amount of MgO required carries with it too much lime, the dolomitic lime may be enriched by adding MgO so as to keep the lime dosage down to that which can be consumed usefully in the reactions. This is sometimes the case in waters that are low in Mg^{++}, HCO_3^-, and CO_2, so that little lime is required for treatment.

However, there is a special problem involved in transforming the undissolved MgO into the ionic Mg. Equations [4] and [5] showed that CO_2 or HCO_3^- must be present to dissolve MgO. It follows also that, so long as appreciable hydroxide is present (high pH), MgO does not dissolve. Such a condition prevails in the usual precipitation process, where sufficient excess hydroxide (usually in the form of lime) is added to complete the precipitation of ionic Mg after neutralizing the HCO_3^- and CO_2.

Thus, in order to dissolve any MgO, it is necessary to transfer it from the high pH lime-precipitation zone (i.e., the lime-softening reaction and settling tank) to an environment of low pH value (i.e., to a tank or compartment that is separate from the high pH zone).

The raw water, which usually contains some HCO_3^- and CO_2, constitutes such a low pH environment. Thus, if MgO is the source of ionic Mg, it is a simple matter to feed this MgO to the raw water (with or without additional CO_2 as outlined previously) in a separate tank or compartment, i.e., the magnesium dissolver.

But if calcined dolomitic lime $[Ca(OH)_2 \cdot MgO]$ is used as the source of Mg, it becomes necessary to separate the MgO from the $Ca(OH)_2$ before the MgO can be dissolved to yield Mg^{++} in solution. This can be done in a lime-saturator type of apparatus, however, that is cumbersome and expensive. A much simpler method is to use the water-treating reaction and settling tank for this separation. The method of doing this is diagrammatically illustrated in Fig. 2.

This simple arrangement makes it possible to use the dolomitic lime as a source of low-cost ionic Mg by withdrawing the

TABLE 1 SOURCES AND COSTS OF MAGNESIUM IN SOLUBLE OR DISSOLVABLE COMPOUNDS

Commercial name	Chemical name	Chemical formula of usual commercial form	Average analysis CaO, per cent	Average analysis MgO, per cent	Cost of material, cents per lb	Value of lime (CaO) content, cent per lb	Balance of cost of material, cents per lb	Number of gram equivalents of MgO in 1 lb of commercial material	Cost per gram equivalent of MgO,[b] cent
Epsom salt	Magnesium sulphate	$MgSO_4 \cdot 7H_2O$	0	16	1.0	0[c]	1.0	3.6	0.53
Calcined magnesite	Magnesium oxide	MgO	1–2	92	3.0	0	3.0	21.0	0.14
Calcined dolomite (hydrated)	Calcium hydrate-magnesium oxide	$Ca(OH)_2 \cdot MgO$	47[a]	33[a]	0.6	0.3	0.3	7.5	0.04
Calcined dolomite (unhydrated)	Calcium magnesium oxide	$CaO \cdot MgO$	57[a]	42[a]	0.6	0.3	0.3	9.5	0.03

[a] These figures correspond to CaO MgO mole ratios = 1:1, which is to be expected from the fact that these materials are produced by calcining natural dolomite ($CaCO_3 \cdot MgCO_3$).

[b] 1 milliequivalent per liter of water (meq per l) corresponds to

$$\frac{1{,}000{,}000 \text{ lb}}{2.2 \text{ lb per l}} = 454{,}000 \text{ meq or } 454 \text{ g eq per million lb of water or } 3.8 \text{ g eq per } 1000 \text{ gal of water.}$$

[c] In fact, $MgSO_4$ requires (in addition to the lime for precipitating the Mg^{++}) Na_2CO_3 for precipitating the $CaSO_4$ left in solution; this is usually a substantial item in operating cost.

MgO-containing thickened sludge from the settling tank after the lime is consumed in the reactions, the MgO of the dolomitic lime having dropped to the bottom of the settling tank with the precipitated $Mg(OH)_2$ and $CaCO_3$. This MgO-containing sludge is then agitated in the low pH environment in the magnesium dissolver, where it meets the HCO_3^-- and CO_2-containing raw water.

It should be noted that, although magnesium in the form of natural magnesite ($MgCO_3$) or dolomite ($CaCO_3 \cdot MgCO_3$) is even cheaper than in the corresponding calcined forms shown in Table 1, it is not practical to use these natural uncalcined materials as sources of Mg, because they cannot be dissolved in the presence of HCO_3^- and CO_2. However, under certain

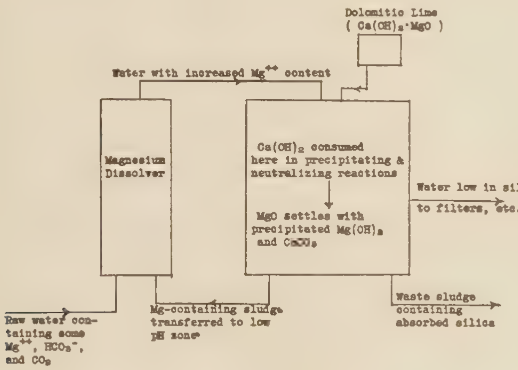


FIG. 2 DOLOMITIC LIME AND RECYCLED SLUDGE AS A SOURCE OF IONIC MG

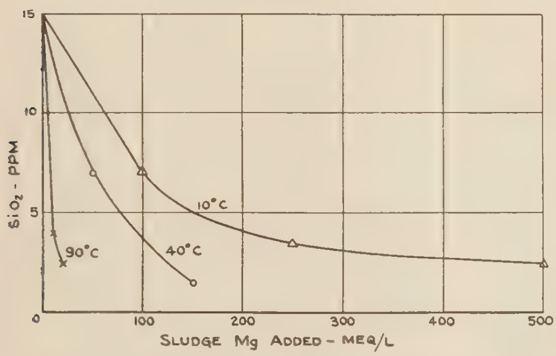


FIG. 3 SILICA REMOVAL BY UNDISSOLVED SPAULDING PRECIPITATOR SLUDGE AT VARIOUS TEMPERATURES

NOTE: (a) Raw well water containing 15 ppm SiO_2.
(b) Spaulding precipitator sludge containing 30 per cent by weight of MgO and $Mg(OH)_2$ and 70 per cent $CaCO_3$.
(c) Agitated with sludge 30 min; then centrifuged 5 min, decanted and filtered through paper, and SiO_2 determined on filtrate.
(d) No ionic Mg precipitated.

conditions, a partially calcined dolomite ($CaCO_3 \cdot MgO$) may be satisfactory and economical.

SILICA REMOVAL BY UNDISSOLVED MAGNESIUM COMPOUNDS (WITHOUT PRECIPITATING IONIC MG)

Exploratory tests made previously showed the advantages of accumulated Mg-containing sludges. It was clear from these observations that undissolved Mg compounds also have silica-absorption properties. This fact, as well as the desirability of avoiding the dissolving and reprecipitation of ionic Mg in some cases, made it necessary to study the role of undissolved Mg compounds.

These investigations included tests on various sludges and

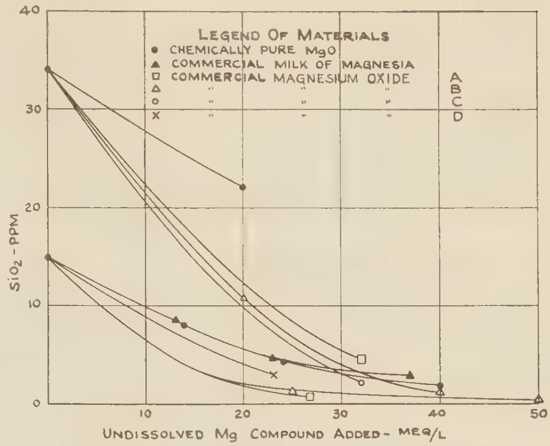


FIG. 4 INITIAL SILICA ABSORPTION BY VARIOUS UNDISSOLVED MG COMPOUNDS

NOTE: (a) Raw well water containing 15 to 16 ppm SiO_2 (lower set of curves) and 32 to 34 ppm SiO_2 (upper set of curves).
(b) Temperature 90 C.
(c) Water agitated 20 min with each material after neutralizing HCO_3^- and CO_2 to prevent MgO from passing into solution. Sample then decanted, filtered through paper, and SiO_2 determined on filtrate.
(d) No ionic Mg precipitated.

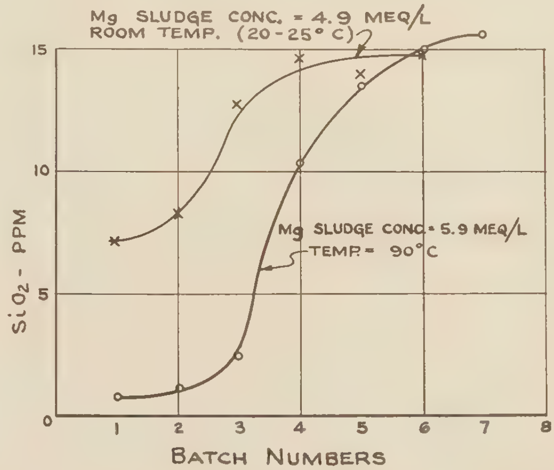


FIG. 5 ULTIMATE SILICA ABSORPTION BY VARIOUS UNDISSOLVED MG COMPOUNDS

NOTE: (a) Raw well water containing 15 to 16 ppm SiO_2.
(b) Successive batches of water treated with one batch of externally prepared $Mg(OH)_2$ sludge [by precipitation from $Mg(HCO_3)_2$ solution] in concentrations indicated on curves.
(c) Water agitated with sludge 1 hr on each test after neutralizing HCO_3^- and CO_2 to prevent $Mg(OH)_2$ from passing into solution. Sample then decanted, filtered through paper, and SiO_2 determined on filtrate.
(d) No ionic Mg precipitated.

various commercial magnesium compounds. Precipitation of ionic Mg was eliminated so as to observe the effect of the undissolved compounds only; for this purpose, a low Mg^{++} water was used, and the HCO_3^- and CO_2 were neutralized before adding the Mg compound, so as to prevent any magnesium from passing into solution. The tests were also extended to various temperature ranges, and the initial and final SiO_2 were measured. The results are set forth in Fig. 3. The tests were conducted on the same well water used for the ionic Mg tests and containing about 15 ppm SiO_2. The sludge was taken from a precipitator which had been operating for some time on this well-water supply; thus the sludge was substantially contaminated with absorbed silica.

These studies show that:

(*a*) The larger the amount of sludge magnesium added, the greater is the silica reduction.

(*b*) The amount of undissolved sludge Mg required to produce a given silica reduction is much greater than the amount of precipitated ionic Mg required.

(*c*) The efficiency of silica reduction increases with rise in temperature in the same general way as with precipitated ionic Mg.

These findings on silica absorption by Mg sludges were then supplemented by determining the silica-absorption characteristics of various other undissolved Mg compounds. The results are summarized in Fig. 4. The term "initial absorption" means the silica reduction obtained by contacting a water once with a given material. Such tests are made in the absence of ionic Mg precipitation or accumulated undissolved Mg compounds.

Since we already had indications of the value of accumulated sludge, further investigation was undertaken to determine more clearly the nature of this reaction. For this purpose, so-called "ultimate-absorption" tests were made as illustrated in Fig. 5. These tests show that, as successive batches of water are brought into contact with these undissolved Mg compounds, they continue to absorb more silica. It follows from this that there is economy to be realized by retaining these partially exhausted materials in the system for further contact with silica-containing water. It would be expected that such further silica absorption by the partially exhausted Mg compound would reduce the silica-absorption burden on new Mg compound added to the system (make-up Mg compound), and would therefore reduce the required quantity of such make-up Mg compound.

In order to check these theoretical conclusions, tests were conducted with various undissolved Mg compounds, and the material added was accumulated from one batch treatment to the next until certain desired concentrations of these accumulated compounds had been reached. These accumulations are referred to as "sludge accumulations," and their concentrations on the graphs are represented in terms of "sludge Mg concentrations—meq/L."

An alternative and more rapid method of accumulating such silica-contaminated sludge is to treat a large volume of water with the Mg compound in question, thereby contaminating the MgO with silica. The large amount of contaminated MgO thus obtained is separated from the water and used as the contaminated sludge for subsequent treatments of small volumes of water.

To these successive batches of water, specified amounts of make-up MgO are added until equilibrium is reached under each set of conditions for at least several batches, and just prior to adding the make-up MgO, a portion of the sludge is discharged to waste in an amount equal to the make-up MgO to be added to the system. Thus the total sludge Mg concentration is maintained at a constant figure for each series of measurements with a specific make-up MgO dosage. This provides a method for collecting equilibrium data which stimulate the results that are obtained in large-scale operation.

Figs. 6 and 7 summarize the results obtained for certain Mg compounds at 95 C and 20 to 25 C, respectively. The absorption of silica by adding make-up undissolved Mg compounds in the presence of Mg sludge accumulations is referred to as "operating absorption." These graphs show clearly that the accumulation of Mg sludges substantially reduces the amount of make-up MgO required for obtaining a given silica-removal result or, conversely, that the same amount of make-up MgO produces a better silica-removal result in the presence of Mg sludge accumulations.

Sources and Desirable Characteristics of Undissolved Mg Compounds

In general, the same materials as those mentioned under sources of ionic Mg are suitable for silica absorption in the undissolved form, calcined dolomite [$Ca(OH)_2 \cdot MgO$] being one of the cheapest sources (see Table 1). Usually it is necessary to add

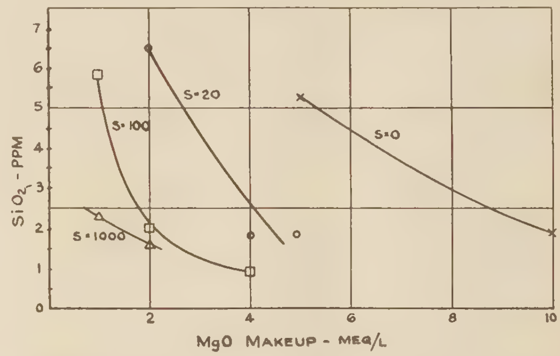


Fig. 6 Effect of Accumulating Mg Sludges on Silica Removal by Undissolved Mg Compounds; High Temperature

Note: (*a*) *S* represents sludge-magnesium concentration in meq/L.
(*b*) Material used as make-up and accumulated as sludge was a commercial calcined magnesite about 300 mesh.
(*c*) Raw well water containing 15 ppm SiO_2.
(*d*) Temperature 95 C.
(*e*) Agitated with sludge 20 min.
(*f*) pH of filtered effluent 10 to 10.3.
(*g*) HCO_3^- and CO_2 first neutralized with lime to prevent any of the Mg from passing through ionic stage.
(*h*) No ionic Mg precipitated.

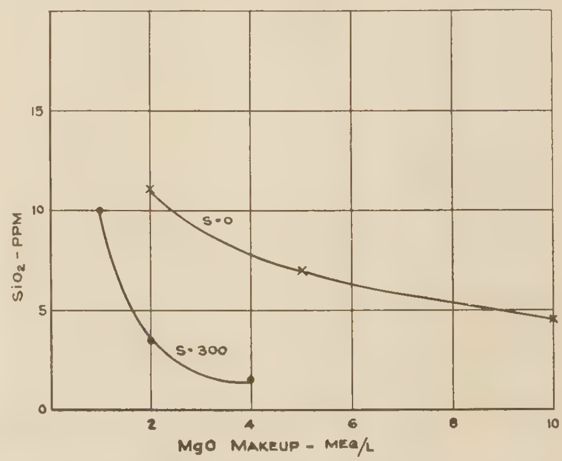


Fig. 7 Effect of Accumulating Mg Sludges on Silica Removal by Undissolved Mg Compounds; Low Temperature

Note: (*a*) *S* represents sludge-magnesium concentration in meq/L.
(*b*) Material used as make-up and accumulated as sludge was a commercial calcined magnesite about 300 mesh.
(*c*) Raw well water containing 15 ppm SiO_2.
(*d*) Temperature 20 to 25 C.
(*e*) Agitated with sludge 1 hr.
(*f*) pH of filtered effluent 10 to 10.3.
(*g*) HCO_3^- and CO_2 first neutralized with lime to prevent any of the Mg from passing through ionic stage.
(*h*) No ionic Mg precipitated.

some lime in the treatment even if Mg^{++} is not being precipitated; as a matter of fact, the HCO_3^- and CO_2 in the water must be neutralized to prevent the MgO from passing into solution. Thus, if dolomitic lime is used, there is added with each equivalent of $Ca(OH)_2$ one equivalent of low-cost MgO. If this is not sufficient, the dolomitic lime may be supplemented by magnesium oxide (MgO).

Tests show that, when dolomitic lime is used in this man-

ner, it is preferable to avoid feeding it in the form of a slurry, because the absorption of OH ion by the MgO from this concentrated-lime solution reduces the silica-absorption capacity of the MgO. (Some of our recent data contradict this statement, but there has not yet been an opportunity to determine the final answer to this question.) Our investigations have shown also that other things being equal, finer particle size enhances the silica absorption by undissolved Mg compounds. Also there is a relation between the calcining temperature of various materials and their ability to absorb silica.

Combined Use of Ionic Magnesium and Undissolved Magnesium Compounds

Although in the preceding discussion and in much of the investigation, these two types of Mg have been separated for purposes of study, in actual practice both forms of Mg are usually used in combination. This is due to the fact that most raw waters contain some Mg^{++}, and, if not, they usually contain some HCO_3^- and CO_2, so that some ionic Mg can be conveniently added by the simple expedient of agitating the raw water with an Mg-containing sludge in the magnesium dissolver (Equations [4] and [5]).

In this connection, it should be noted that, if ionic Mg is added to the water by dissolving MgO or $Mg(OH)_2$ and if no CO_2 is added to the system, the amount of lime required for the precipitation of this Mg^{++} is the same as that which would have been required for reacting with the $CO_2 + HCO_3^-$ in the usual lime treatment. This is apparent from a consideration of the following reactions, which take place in the usual lime treatment.

$$CO_2 + Ca(OH)_2 \rightarrow CaCO_3 + H_2O \qquad [6]$$

$$2HCO_3^- + Ca(OH)_2 \rightarrow CaCO_3 + CO_3^- + 2H_2O \qquad [7]$$

In each of these reactions, 1 equivalent of lime is consumed, while according to Equations [4] and [5], the reactions between MgO and the same amounts of CO_2 and HCO_3^- yield one equivalent of $MgCO_3$ in each case. As shown by Equation [2], each of these equivalents of $MgCO_3$ in turn requires only 1 equivalent of $Ca(OH)_2$, i.e., the same amount as required for the reactions directly with lime as set forth in Equations [6] and [7]. The OH ion liberated by the hydrolysis of the MgO or directly by the $Mg(OH)_2$ exactly balances the OH ion required in the form of lime (or other alkali) for the precipitation of the dissolved $MgCO_3$. It follows that the mere addition of ionic Mg, dissolved from MgO by the $CO_2 + HCO_3^-$ present in the raw water, does not cause any increase in the lime consumption. Also it is clear that the amount of Mg^{++} picked up in the magnesium dissolver does not alter the lime dosage in any way, i.e., the hardness and alkalinity of the effluent remain constant.

Where conditions require the precipitation of more ionic Mg than can be directly taken up by the raw water, the CO_2 content of the water is increased by means of flue gas as discussed previously. In that event, each added equivalent of CO_2 requires the addition of one equivalent of lime. Usually this does not increase the operating cost out of proportion to the added benefits obtained from the high Mg^{++}.

Silica Contamination of Sludge Accumulations

In practicing the combined ionic Mg and undissolved Mg process, the Mg^{++} which is precipitated as $Mg(OH)_2$ concentrates in the sludge zone of the settling tank with the MgO of the dolomite. During this cycle, the $Mg(OH)_2$ and MgO are contaminated with the silica which they absorb, and such contamination would increase indefinitely were it not for the fact that there is discharged to waste from the system a certain amount of silica-contaminated sludge, while there is being added to the system fresh Mg. If the Mg added to the system and wasted is increased, this ratio of $Mg:SiO_2$ in the sludge rises correspondingly, and the residual SiO_2 in the effluent decreases, Fig. 6. Therefore, in calculating the amount of make-up MgO to be added in the form of $Ca(OH)_2 \cdot MgO$ or MgO, this contamination factor must be considered, i.e., the amount of make-up MgO must bear a certain ratio to the SiO_2.

Spaulding Precipitator-Type Equipment Favorable for Process

In addition to the laboratory-scale tests on both the ionic Mg, the undissolved Mg compounds, and the combined process, extended tests were carried out on a miniature-plant scale using various combinations of ionic and undissolved Mg, at various temperatures, and with various sludge concentrations. These extensive tests indicate that the most favorable type of equipment is that illustrated in Fig. 8. The principles of the precipitator itself have been described (47), and it is clear that this equipment has certain characteristics which make it particularly suitable for carrying out this process.

For example, as shown in Fig. 8, the magnesium dissolver can be conveniently built into the upper section of the downcomer without any sacrifice of useful space. Also, this type of equipment lends itself readily to the insertion of so-called sludge-concentrator pockets in which the sludge is further thickened. This thickened sludge is especially suitable for recirculation into the

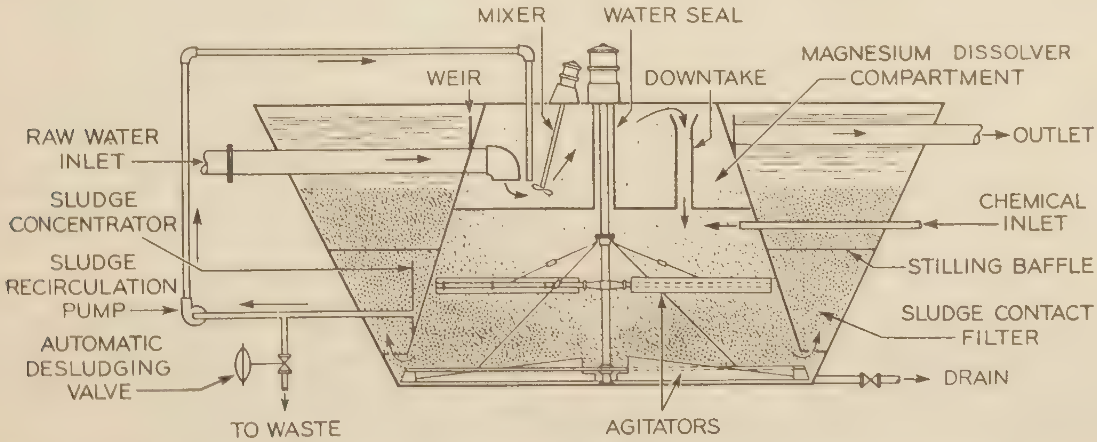


Fig. 8 Modified Spaulding Precipitator With Magnesium Dissolver, Etc., for Magnesia-Silica-Removal Process

magnesium dissolver, because the higher the sludge concentration in this recirculated stream, the smaller is the volume required to maintain the proper MgO concentration in the magnesium dissolver. Also, it is desirable to add as little as possible of the high pH liquid to the magnesium dissolver, because any OH^- or $CO_3^=$ in the recirculated liquid neutralizes some of the HCO_3^- and CO_2 in the raw water, and thereby reduces the amount of Mg^{++} that can be dissolved in the magnesium dissolver.

As shown previously, thorough and extended agitation of the concentrated sludge with the water enhances the silica-removal process. It is clear that the Spaulding precipitator affords this extended intimate contact between the water and the sludge, which is maintained in suspension in the lower part of the equipment. Briefly, the operation, illustrated in Fig. 8, is as follows: The raw water and recirculated sludge are mixed in the magnesium-dissolver compartment, where the water takes up the required amount of Mg^{++}. This water then passes into the downcomer, where it meets the stream of chemical reagents and undergoes thorough agitation for an extended period of time, during which the Mg^{++} is precipitated as $Mg(OH)_2$ and the Ca^{++} is precipitated as $CaCO_3$. The water containing these precipitates and the undissolved MgO from the dolomitic lime or other source then passes through the ports adjacent to the lower edge of the central downcomer and passes upward through the concentrated suspension of previously formed sludge in the outer compartment. As this compartment increases in cross-sectional area, the vertical velocity of the water continuously decreases as it rises toward the outlet. Finally, the upward velocity of the water decreases to such a point that it can no longer maintain the sludge particles in suspension. At this level, the sludge is left behind, and the clear water rises to the top of the equipment and overflows the collecting weir and passes through the outlet and into the filters.

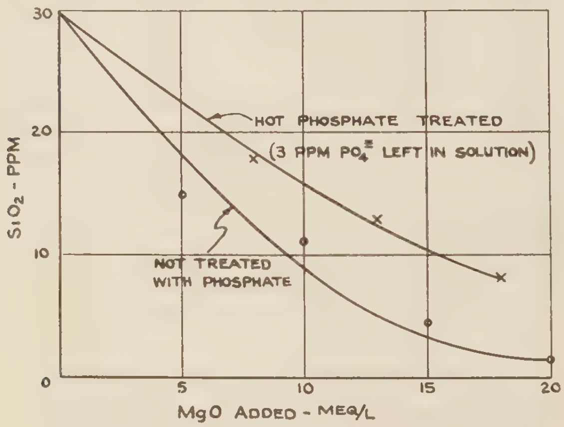


Fig. 9 Effect of Phosphate on Initial Silica-Absorption Capacity

Note: (a) Raw water containing 30 to 33 ppm SiO_2.
(b) Commercial calcined magnesite (finer than 200 mesh) added without any sludge accumulations.
(c) Temperature 95 C.
(d) Agitated with MgO 20 min; then decanted and filtered, and SiO_2 determined on filtrate.
(e) HCO_3^- and CO_2 neutralized on untreated sample to prevent MgO passing into solution.
(f) No ionic Mg precipitated.

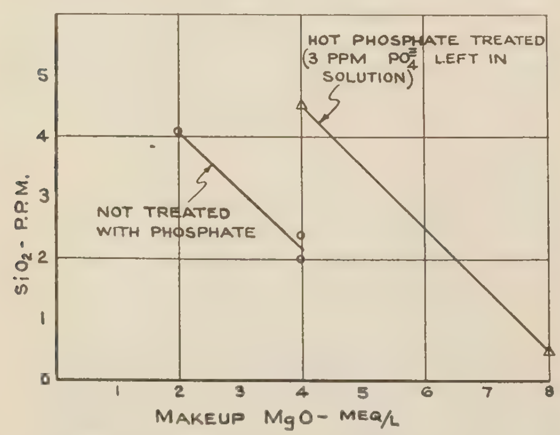


Fig. 10 Effect of Phosphate on Operating Silica-Absorption Capacity in Presence of Concentrated Sludge Accumulations

Note: (a) Raw water containing 30 to 33 ppm SiO_2.
(b) Commercial calcined magnesite (finer than 200 mesh) added in presence of 100 meq sludge Mg per l.
(c) Temperature 95 C.
(d) Agitated with MgO 20 min; then decanted and filtered, and SiO_2 determined on filtrate.
(e) HCO_3^- and CO_2 neutralized on untreated sample to prevent MgO passing into solution.
(f) No ionic Mg precipitated.

In addition to accomplishing the silica removal by the precipitation of Mg^{++} and by maintaining intimate contact between the sludge and the water, this equipment permits the usual lime-soda softening reactions to take place at the same time. Also the addition of coagulant provides for simultaneous turbidity removal. It is evident, therefore, that this silica-removal process can be incorporated in the lime-soda-coagulant-softening and turbidity-removal plant at little added initial cost or operating cost.

The process can also be incorporated into the usual hot-process lime-soda softener. For the most efficient results, it is desirable to incorporate the Spaulding sludge-agitation and sludge-contact features (and occasionally the magnesium-dissolver feature) in such equipment. With this in view, the construction principles, illustrated in Fig. 8, are being incorporated in a modified hot-process lime-soda softener which is under construction at the present time.

Effect of Phosphate on Silica Removal

A question arises as to the effect of phosphates on this silica-removal process. In some cases where the raw-water hardness is very low, it is found desirable to soften the feedwater make-up by external hot-phosphate treatment, which is carried out in equipment similar to the conventional hot-process lime-soda softener. Since many such low-hardness waters have a high silica content, the effect of phosphate ion on the absorption of silica by undissolved MgO was investigated. Wesly (13, 42) reported that phosphate ion interfered with silica absorption by undissolved MgO, and our tests were made to determine the extent of this interference under conditions prevailing in the present process.

Two series of tests were carried out; one on the so-called initial silica absorption, where the MgO is contacted once with the water without sludge accumulations, and a second on the so-called operating silica absorption, which is carried out in the presence of concentrated accumulations of Mg compounds or sludge and only a certain amount of make-up Mg compound is added to the system. In each case, the results were compared on the hot-phosphate-treated water with those on a water which was not treated with phosphate.

Figs. 9 and 10 summarize the data. They show clearly that the presence of phosphate ion interferes substantially with the silica-absorption properties of MgO. It may be concluded, therefore, that in such cases it is usually preferable to apply the silica-removal process before applying the hot-phosphate treatment.

Conclusions

This research study has revealed the following:

(*a*) For any given set of conditions ionic Mg precipitated in situ as $Mg(OH)_2$ is most efficient, but undissolved Mg compounds of proper quality (fine-particle size, etc.) also give satisfactory results when favorable conditions of sludge concentration, agitation, etc., are maintained.

(*b*) The efficiency of removing silica by precipitating ionic Mg or by undissolved Mg compounds increases with temperature, but the application of other factors disclosed makes it possible to obtain satisfactory results at low temperature. Slight prewarming of the water to about 30 to 40 C (86 to 104 F) reduces the Mg required to obtain the desired result; waste heat is usually available for this purpose.

(*c*) Efficiency increases with the concentration of the Mg-containing sludge in contact with the water.

(*d*) Increasing the time of contact with the sludge absorbent in agitated suspension improves the results.

(*e*) Reducing the silica contamination of the retained concentrated sludge reduces the silica in the effluent: thus if the Mg added to the system is increased, the ratio of $Mg:SiO_2$ in the sludge rises correspondingly, and the residual SiO_2 in the effluent decreases.

(*f*) The presence of phosphate ion interferes with silica absorption by MgO.

(*g*) In those cases where additional Mg is required for the desired silica reduction, it can usually be obtained most economically from dolomitic lime ($Ca(OH)_2 \cdot MgO$); where no lime is required in the treatment, calcined magnesite (MgO) is used.

(*h*) Ionic Mg may be added to the raw water economically and without increasing the lime requirement or the total-solids content of the effluent by recirculating sludge into a magnesium-dissolver compartment or low pH zone, where the HCO_3^- and CO_2 in the raw water dissolves Mg^{++} from this sludge. If necessary, additional CO_2 may be added by flue-gas carbonation to increase further the amount of Mg^{++} introduced.

In applying this silica-removal process, the economic balancing of these findings to produce the desired result requires careful consideration of all the facts in any given case. The data show that the silica may be removed efficiently both at low and high temperatures. In general, the following may be stated with respect to the equipment:

(*a*) If hot-process lime-soda treatment is applied to reduce the hardness, superior silica-removal results are obtained by using equipment embodying positive agitation and intimate sludge contact like that provided by a Spaulding precipitator type of hot-process softener.

(*b*) If hot-phosphate softening is used, the silica-removal treatment should be carried out first, and the phosphate treatment should follow.

(*c*) If the treatment is applied at low or medium temperatures (cold-process treatment), the silica removal is applied first, and advantage may be taken of posttreatment by carbonaceous zeolites, which completely eliminate any residual hardness. Carbonaceous hydrogen-zeolite treatment (14) may also be included to reduce the final alkalinity to any predetermined figure. These carbonaceous zeolites, being nonsiliceous, can treat the effluent of the silica-removal plant without danger of silica pickup from the zeolite by the water.

Acknowledgment

The author wishes to acknowledge the cooperation of Mr. S. B. Applebaum, vice-president of The Permutit Company and of the following members of The Permutit Company's Research and Development staff in the development of this process and the preparation of this paper: Dr. P. C. Goetz, Dr. John G. Dean, Dr. Ray Riley, Mr. R. W. Jackson, Dr. Calvin Calmon, Mr. M. E. Gilwood, and several others.

BIBLIOGRAPHY

1 "Stellungnahme zu dem Bericht von Oberingenieur Pfadt," by K. Morawe, Vereinigung der Grosskesselbesitzer, Speisewasserpflege, Berlin, 1928, pp. 44–47.

2 "Untersuchungen über die Entkieselung des Wassers," by R. Stumper, *Die Wärme*, vol. 55, 1932, pp. 272–278.

3 "Zur Kenntnis der zementierenden, insbesondere der silicatischen Bestandteile des Kesselsteines," by F. Hundeshagen, *Chemiker Zeitung*, vol. 56, 1932, pp. 521–524, 542–544.

4 "Silicon in Boiler Waters," by R. E. Summers and C. S. Keevil, unpublished paper, presented at Annual Meeting, A.S.M.E., 1933; *Power Plant Engineering*, vol. 38, 1934, pp. 219–220.

5 "Treatment of Silica for High Pressure Boilers," by S. E. Tray, *Power Plant Engineering*, vol. 40, 1936, pp. 515 517.

6 "Complex Silicate Scales in High-Pressure Boilers," by S. T. Powell, L. V. Carpenter, and J. J. Coates, *Combustion*, vol. 11, no. 2, August, 1939, pp. 18–24.

7 "Die Entkieselung des Speisewassers der Höchstdruckanlage in Höchst," by W. Geisler, *Mitteilungen der Vereinigung der Grosskesselbesitzer*, vol. 64, 1937, pp. 267–269.

8 Discussion at Mechanical Engineers Meeting in Lone Star State, *Power*, vol. 80, 1936, p. 381.

9 "Zur Verkieselung der Dampfturbinen," by F. Kaissling, *Archiv für Wärmewirtschaft und Dampfkesselwesen*, vol. 20, 1939, pp. 89–91.

10 Abstract of paper by W. Wesly, *Die Wärme*, vol. 61, 1938, p. 576; refer to (13).

11 "Descaling Steam Turbines," by J. B. Entriken, *Industrial and Engineering Chemistry*, vol. 30, 1938, pp. 1279–1280.

12 "Die Kieselsäure und ihre Bedeutung im Hochdruck-Kesselbetrieb," by A. Splittgerber, *Mitteilungen der Vereinigung der Grosskesselbesitzer*, vol. 73, 1939, pp. 206–231.

13 "Neue Erfahrungen über die Speisung von Höchstdruckkesseln mit chemische aufbereitetem Wasser," by W. Wesly, *Chemische Fabrik*, vol. 12, 1939, pp. 137–142.

14 "Carbonaceous Zeolites—an Advance in Boiler-Feedwater Conditioning," by H. L. Tiger, Trans. A.S.M.E., vol. 60, 1938, pp. 315–325.

15 "Applications of Carbonaceous Zeolites to Water Softening," by S. B. Applebaum, *Journal*, American Water Works Assn., vol. 30, 1938, pp. 947–978.

16 "Treatment of Boiler Feedwater by Zeo-Karb Process," by J. D. Yoder, *Combustion*, vol. 10, no. 11, 1939, pp. 35–40.

17 "Zeo-Karb H: A New Method of Conditioning Water to Remove Sodium Bicarbonate," by S. B. Applebaum and R. Riley, *Industrial and Engineering Chemistry*, vol. 30, 1938, pp. 80–82.

18 "Standard Methods for the Examination of Water and Sewage," American Public Health Association, Eighth edition, New York, N. Y., 1936, pp. 103–105.

19 "Colorimetric Standards for Silica," by H. W. Swank and M. G. Mellon, *Industrial and Engineering Chemistry*, Analytical edition, vol. 6, 1934, pp. 348–350.

20 "Silicomolybdate Method for Silica," by H. W. Knudson, C. Juday, and V. W. Meloche, *Industrial and Engineering Chemistry*, Analytical edition, vol. 12, 1940, pp. 270–273.

21 "Silica in Water," by M. C. Schwartz, Louisiana State University Bulletin, vol. 30, N.S. no. 14, Oct., 1938.

22 "Determination of Soluble Silica in Water. A Photoelectric Method," by H. L. Kahler, presented at the 100th Meeting of the American Chemical Society, September 11, 1940.

23 "Precipitation of Silica in Drinking Water by Means of Soda and Alum," by A. Massink, Verslag Centr. Lab. t.b.u.h., Staatstaez. o.d. Volksgesend. h. over, 1918, pp. 76–83.

24 "Silicic Acid, Its Influence and Removal in Water Purification," by O. M. Smith, Illinois State Water Survey, Bulletin 16, 1918-1919, pp. 140–181.

25 "Über die Entkieselung von Kieselsäurehaltigen Wässern," by E. Berl and H. Staudinger, *Zeitschrift für Angewandte Chemie*, vol. 40, 1927, pp. 1313–1317.

26 "Removing Silica From Water," by Otto Liebknecht, U. S. Pat. 1,860,781, May 31, 1932, assigned to The Permutit Company.

27 "Verfahren zur Entfernung von in Wasser gelöster Kieselsäure," by A. G. Kurz, Austrian Pat. 133,757, June 15, 1933.

28 "Prevention of Silica Scale With Sodium Aluminate," by C. H. Christman, J. A. Holmes, and H. Thompson, *Industrial and Engineering Chemistry*, vol. 23, 1931, pp. 637–646.

29 "Kolloidchemische Fragen der Speisewasserbehandlung, . . ." by R. Stumper, *Die Wärme*, vol. 53, 1930, pp. 859–862.

30 "Coagulation as Applied to Industrial Waters," by A. P. Black, R. A. Bardwell, and B. W. Graham, *Journal*, American Water Works Assn., vol. 29, 1937, pp. 1533–1547.

31 "Removal of Silica From Water by Sodium Aluminate," by F. K. Lindsay and J. W. Ryznar, *Industrial and Engineering Chemistry*, vol. 31, 1939, pp. 859–861.

32 "Water Treatment," by F. K. Lindsay and D. G. Braithwaite, U. S. Pat. 2,194,524, March 26, 1940, assigned to National Aluminate Corporation.

33 "Removal of Silica From Water by Cold Process," by L. D. Betz, C. A. Noll, and J. J. Maguire, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 1320–1323.

34 "The Removal of Silica From Water for Boiler Feed Purposes," by M. C. Schwartz, *Journal*, American Water Works Assn., vol. 30, 1938, pp. 659–678.

35 "Rohwasser-Entkieselung," by K. Hoell, *Archiv für Wärmewirtschaft*, vol. 19, 1938, pp. 323–324.

36 "Recent Trends in Water Treatment," by S. T. Powell, L. V. Carpenter, L. R. Setter, and J. J. Coates, *Chemical and Metallurgical Engineering*, vol. 46, 1939, pp. 481–484.

37 "Removal of Silica From Water," by A. S. Behrman and H. Gustafson, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 468–472.

38 "Process for Purifying Water," by H. A. Reimers, U. S. Pat. 1,966,733, July 17, 1934, assigned to Dow Chemical Company.

39 "Verfahren zum Reinigen von Wasser unter Anwendung von Magnesiumoxyd oder basisch Kohlensaurer Magnesia," by E. Bohlig, German Pat. 3187, Jan. 16, 1879.

40 "Method of Removing Dissolved Silica From Natural Alkaline Brines," by W. A. Kuhnert, U. S. Pat. 1,992,532, Feb. 26, 1935.

41 "Removal of Silica From Solution at Boiler Temperatures," by F. G. Straub, *Industrial and Engineering Chemistry*, vol. 28, 1936, pp. 36–37.

42 Discussion of paper by W. Geisler (7), by W. Wesly, *Mitteilungen der Vereinigung der Grosskesselbesitzer*, vol. 64, 1937.

43 "Silica in Boiler Feed Water," by W. F. Peters, Jr., and W. D. Turner, *Power Plant Engineering*, vol. 44, no. 3, 1940, pp. 54–57; no. 4, 1940, pp. 47–49.

44 "Removal of Silica From Water by Hot Process," by L. D. Betz, C. A. Noll, and J. J. Maguire, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 1323–1329.

45 "Adsorption of Soluble Silica From Water," by L. D. Betz, C. A. Noll, and J. J. Maguire, *Industrial and Engineering Chemistry*, June, 1941, pp. 814–820.

46 "Quality of Lime Desired for Water Treatment," by C. P. Hoover, A.S.T.M. Symposium on Lime, March 8, 1939, pp. 43–46.

47 "Cold Lime-Soda Water Softening," by S. B. Applebaum, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 678–684.

Discussion

L. D. Betz,[4] C. A. Noll,[5] and J. J. Maguire.[6] This paper presents data which serves to confirm some of the conclusions previously reached by the writers.[7] We have previously pointed out the increase in efficiency of silica removal, which results from increase in temperature, increase in retention and recirculation of sludge, and are naturally pleased that independent investigators have confirmed the validity of our former conclusions.

Unlike the author, however, we have not found ionic magnesium precipitated in situ as $Mg(OH)_2$ to be more efficient in the removal of silica from solution than the use of properly prepared magnesium oxide, except within the limits of possible error in laboratory technique. Under full-scale-plant conditions, a study of the results obtained have indicated very little, if any, difference in the quantity of silica removed, per unit of magnesium added, whether in the dissolved form as a salt such as magnesium sulphate or from an undissolved compound such as a specially prepared form of magnesium oxide. It is evident that the author's conclusions on the greater efficiency of ionic magnesium were obtained by the comparison of a soluble salt, such as magnesium sulphate with an "undissolved compound," such as calcined magnesite. In the large amount of work we have done on silica removal, we have tested and reported on a great number of various types of calcined magnesite and have found the efficiency of numerous samples submitted to vary considerably. A special form of magnesium oxide, prepared from sea-water bitterns, has been found to possess qualities markedly superior to calcined magnesite for the removal of silica from water.

Under the heading, "Sources of Undissolved Magnesium Compounds," the author calls attention to the fact that "finer particle size enhances the silica absorption by undissolved magnesium compounds. Also there is a relation between the calcining temperature of various materials and their ability to absorb silica." This confirms work done in our laboratories quite some time ago and was the direct cause of cooperative research with the California Chemical Company, which led to the development, almost two years ago, of a highly efficient magnesium oxide prepared under controlled conditions. It is this specially prepared form of magnesium oxide which they have termed Remosil and which we have found to possess considerably superior silica-removal properties to any other form of magnesium oxide we have tested. It is for this reason they have specifically branded this to differentiate it from calcined magnesite. Such data, we have presented in our previous papers on this subject.

Examination of the data, shown in Fig. 4 of the paper, for the initial silica absorption by various undissolved magnesium compounds, indicates a rather low silica-removal efficiency (parts of silica removed per part reagent employed). For example, the work done on the raw water containing 32 to 34 ppm silica, the results illustrated for commercial calcined magnesite *A*, indicate a reduction of the silica content to approximately 5 ppm, employing approximately 32 meq per l of commercial calcined magnesite *A*. This is equivalent to the removal of 28 ppm of silica by 640 ppm calcined magnesite *A* or a silica removal of 0.044 per part of calcined magnesite. For commercial calcined magnesite *C*, silica reduction was effected down to approximately 3 ppm with the use of approximately 32 meq per l of calcined magnesite *C*, or 30 ppm of silica removed by 640 ppm of calcined magnesite. This is equivalent to 0.047 part of silica removed per part of calcined magnesite *C*.

Reference to Table 2, in a current paper[8] by the writers, will show that silica was reduced from 37.1 ppm to 3.6 ppm with the use of 200 ppm magnesium oxide. This shows the removal of 0.17 part of silica per part of magnesium oxide used. In another case, in Table 3 of the same paper, silica was reduced from 31 ppm to 3 ppm with the use of 150 ppm magnesium oxide. This shows a removal of 0.19 part of silica per part of magnesium oxide used. This information is summarized in Table 2 of this discussion. This represents a silica-removal efficiency four times as great for the specially prepared form of magnesium oxide we used in our work as was secured by the author, with the calcined magnesite which he employed. This checks quite closely with the results the writers obtained in their work, as shown in Fig. 9 of our current paper.[8]

Relatively low silica removal per part of calcined magnesite is also shown by the $S = 0$ curve of Figs. 6 and 7. Fig. 9 of the

[4] General Manager, W. H. & L. D. Betz, Philadelphia, Pa. Mem. A.S.M.E.

[5] Chief Chemist, W. H. & L. D. Betz.

[6] Director, Technical Division, W. H. & L. D. Betz.

[7] "Removal of Silica From Water by Hot Process," by L. D. Betz, C. A. Noll, and J. J. Maguire, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 1323–1329.

"Removal of Silica From Water by Cold Process," by L. D. Betz, C. A. Noll, and J. J. Maguire, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 1320–1323.

"Adsorption of Soluble Silica From Water," by L. D. Betz, C. A. Noll, and J. J. Maguire, *Industrial and Engineering Chemistry*, June, 1941, pp. 814–820.

"Removal of Soluble Silica From Water by Adsorption," by L. D. Betz, presented at Water Conference, Engineers' Society of Western Pennsylvania, November 13, 1940.

[8] "Adsorption Process for Removal of Soluble Silica From Water," by L. D. Betz, C. A. Noll, and J. J. Maguire, Trans. A.S.M.E., vol. 63, 1941, pp. 713–720.

TABLE 2 DATA ON REMOVAL OF SILICA BY USE OF MAGNESIUM OXIDE

	Silica in original water, ppm	Residual silica in treated water, ppm	Silica removed, ppm	Reagent employed, meq per l	Reagent employed, ppm	Silica removed per part of reagent employed
Calcined magnesite *A* (Fig. 4)	32–34[a]	5	28	32	640	0.044
Calcined magnesite *C* (Fig. 4)	32–34[a]	3	30	32	640	0.047
Remosil, MgO (Table 2)	37.1 [b]	3.6	33.5	10	200	0.170
Remosil, MgO (Table 3)	31.0 [b]	3	28	7.5	150	0.190
Remosil, MgO (Table 4)	48.0 [b]	2.5	35.5	13	260	0.175

[a] Reported in author's paper.
[b] Reported by writers.[8]

same paper[8] also shows low silica removal per unit of calcined magnesite. In general, the silica-removal efficiencies obtained check our previous observations that the various forms of calcined magnesite are considerably less efficient for the removal of silica, than the specially prepared highly adsorptive grade of magnesium oxide prepared from sea-water bitterns. Undoubtedly, the low removal efficiencies of the calcined magnesites employed in this work have led the author to the conclusions that ionic magnesium is considerably more effective in the removal of silica than magnesium added in the form of an undissolved magnesium compound. Where the magnesium oxide is added as calcined magnesite, this is true. However, the more adsorptive forms of magnesium oxide, with which most of our work was carried out, possess silica-removal properties per unit of magnesium, comparable with those possessed by soluble magnesium salts.

We have found that it is rather important to control the pH value of the silica-removal process at approximately 10.1. The author, in this paper, makes no mention of any control on the basis of pH.

We have not found it necessary in our experience to employ such a unit as the "magnesium dissolver." It is possible that such a device has been employed in this work because of the relatively low efficiency of the calcined magnesite used. Where a magnesium oxide of a higher degree of efficiency is employed, since there is no appreciable difference in the silica-removal properties of that magnesium oxide, when compared with soluble magnesium salts, the need for a dissolver is eliminated.

We have employed, with very successful results, the simple method of adding the specially prepared magnesium oxide directly to the chemical mixing tank of a softener, at the same time as the lime and soda ash employed for softening.

In this paper, the author, without reservation, concludes (*f*): "The presence of phosphate ion interferes with silica absorption by MgO." This conclusion is based on data shown in Figs. 9 and 10 of the paper. Our work with magnesium oxide is not in agreement with this conclusion. On the basis of laboratory work, with water containing about 30 ppm silica, 3 ppm phosphate ion in a treated water does not interfere with subsequent silica removal by magnesium oxide, within the degree of analytical accuracy.

T. E. Crossan.[9] This paper on silica comes at an opportune time when many high-pressure stations are giving serious consideration to the maintenance of low dissolved silica in boiler water without entailing excessive cost and blowdown. For over a year and a half, silica removal by dolomitic oxide, fed along with quicklime, has been accomplished in the double-cone precipitators at Louisiana Station of the Gulf States Utilities Company. Softening and removal of turbidity from the Mississippi River water are accomplished along with the removal of silica, but no recirculation of sludge is practiced.

Our experience to date, with a few exceptions, has agreed with the conclusions reached by the author. Increased ionic magnesium in the river water, higher temperature, higher sludge concentration in the agitation chamber, and longer agitation time, all aid toward the removal of dissolved silica.

There are, however, certain unfavorable aspects which we have encountered. The introduction of a coagulant (in this case ferric sulphate) definitely retards the silica reduction. During periods of river-water turbidities of over 800 ppm, the removal of silica becomes less effective. In some cases, when the incoming dissolved silica is under 4.5 ppm, and the river-water turbidity is high, no silica reduction resulted. With high dissolved silica and high turbidity in the river water good removal has been obtained.

For a constant percentage of total sludge in the agitation chamber of the precipitator, the percentage of magnesium sludge decreases to small proportions as the river-water turbidity increases. Thus, the silica removal is impaired as the proportion of magnesium in the sludge decreases. Furthermore, if this sludge is recirculated, considerable recirculation must be done before any appreciable magnesium can be dissolved in the incoming water. Recirculation of this sludge to this extent in an already highly turbid water would soon overload the sludge concentrators, and the tank effluent would soon be found to carry sludge. Recirculation of sludge, during periods of high turbidity of the inlet water, therefore, may be impractical. On the other hand, the decrease in silica removal may not entirely be one of decrease in magnesium concentration. It may be due to interference with the surface activity or adsorption properties of the magnesium by the finely divided clay.

Because of the constantly varying conditions of our inlet water, and the rather low river-water-dissolved silica, it is difficult to determine the effect of feeding a dolomitic-oxide slurry. In general, however, the dolomitic oxide (on the basis of equivalent magnesium) appears to be at least one half as effective as ionic magnesium. With entering dissolved-silica contents of over 4.5 ppm, even better results are obtained.

In a system such as ours, the water entering the zeolites must be reduced to 7 pH. For good silica removal with magnesium, higher pH values must be carried to precipitate the magnesium than would be required if only a maximum of calcium-hardness removal were required. This means a higher lime dose for silica removal and, eventually, before the zeolites, a higher sulphuric-acid dose to neutralize it than during periods when silica reduction is not required. Silica removal by magnesium therefore can be accompanied by a higher dissolved solids to the boilers than is experienced during normal operation without silica removal.

Because of the longer detention periods, and the greater sludge concentrations required for removal of silica in the cold with magnesium than is required for softening and clarification alone, a somewhat greater plant investment is required. This is especially so with waters having over 200 ppm turbidity.

When feeding dolomitic oxide along with the quicklime, practically no magnesium goes into solution. Even though this results in less effective silica removal, it does not subject the system to sudden fluctuations in hardness to the zeolites and thus endanger their being overrun, sending excessive hardness to the boilers. If conditions are not correct for silica removal, practically no hardness fluctuation results.

On the other hand, if the sludge-recirculation system of the author's company is used, the amount of magnesium in the sludge, the strength of the sludge, and the quantity of CO_2 must

[9] Superintendent of Production, Gulf States Utilities Company, Baton Rouge, La. Mem. A.S.M.E.

be controlled accurately at all times. Deficiency of CO_2 will result in high alkalinities and high calcium hardness. On the other hand too much CO_2 may result in high magnesium hardness, perhaps high calcium hardness, ineffective coagulation, and the sludge in the unit being carried over with the delivered water. With an unchanging water supply and with an unchanging rate to be treated, the system has the additional need for accurate CO_2 control. However, with a variable water supply and a changing rate, the system will, as we now see it, require extremely careful control and observation.

One of the most important economical factors in the system, described by the author, is the pH of the water entering the downtake, Fig. 8 of the paper. It would be interesting to know the proportions and the amounts of magnesium and calcium dissolved by CO_2 at different pH values. If adequate dissolving of magnesium can be accomplished at 9.1 pH, without too much simultaneous dissolving of calcium, then considerable lime and CO_2 can be saved. At 7.5 pH both magnesium and calcium enter solution very readily, but more CO_2 is required to convert magnesium and calcium to the bicarbonate form, and considerable calcium is dissolved with the magnesium. The lime dose required to reprecipitate the redissolved calcium and magnesium becomes quite large. Therefore, the pH at which redissolving of magnesium takes place would seem to be highly important from an economical standpoint.

The maintenance of low dissolved silica in boiler waters, without high blowdown or excessive chemical cost, would seem to be within reasonable reach of many high-pressure stations, as a result of the comprehensive treatment of the subject in this paper.

C. E. Joos.[10] This paper treats comprehensively the various factors affecting the silica-removal process by magnesium salts. While the author discusses both the cold and hot treatments,

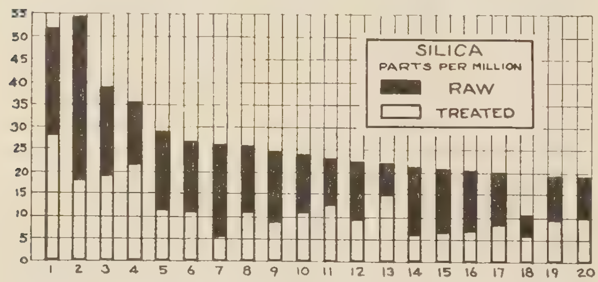


Fig. 11 Reduction in Silica by Use of Hot-Process Softener

the writer wishes to confine his comments to the hot-process treatment for silica reduction when combined with softening.

It has been recognized for many years that the hot-process softener, utilizing lime and soda ash in the treatment of fairly hard water supplies, resulted in considerable silica reduction. Fig. 11, of this discussion, illustrates a series of tests, made on raw and treated waters from hot-process-softener installations, showing the reduction in silica which took place during the softening process without regard to silica reduction as the principal objective. It was also noted that, the higher the magnesium hardness, the more effective was the silica removal, and that calcium precipitants were of slight value. These observations led to the process of fortifying the water with magnesium sulphate wherever the silica reduction was to be greater than that which could be obtained through the precipitation of the natural magnesium hardness of the raw water.

[10] Cochrane Corporation, Philadelphia, Pa.

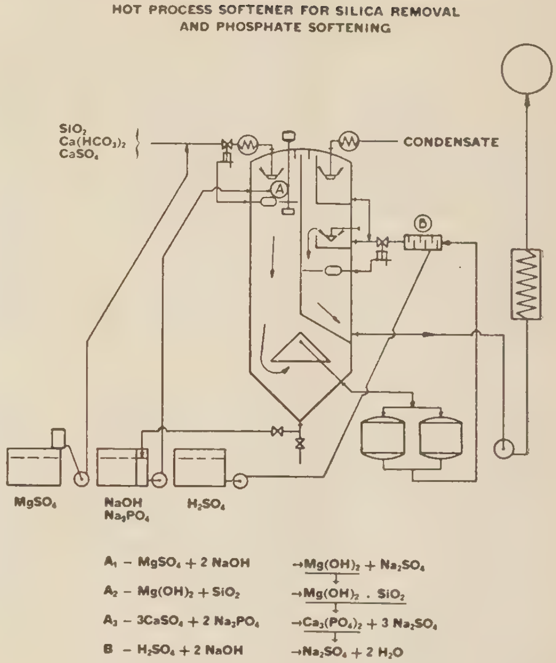


Fig. 12 Hot-Process Softener for Silica Removal and Phosphate Softening

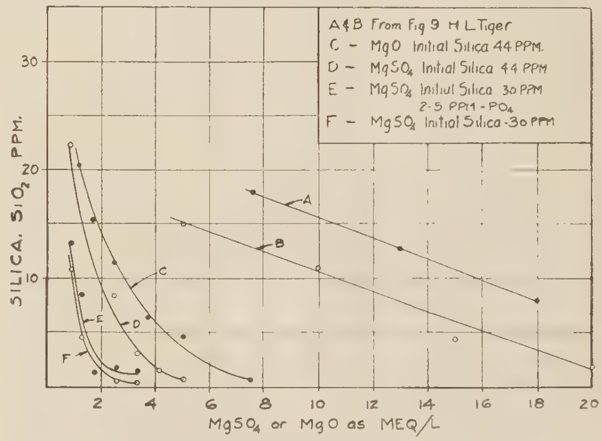


Fig. 13 Comparison of Results in Reduction of Silica by Use of Magnesium Salts

Recommendations of this procedure were made as early as 1936, and a hot-process softener has been in operation for well over a year, specifically installed to reduce silica by the use of magnesium sulphate. Fig. 12, of this discussion, is a diagrammatic outline of this installation. Phosphate is used in this case as the softening agent, because of the low hardness of the raw water. Sludge is recirculated to reduce the consumption of magnesium sulphate and caustic soda.

During our research in the reduction of silica by the use of magnesium salts, it was recognized that magnesium sulphate, although very effective, was open to the objection that it produces a by-product of sodium sulphate which in some cases was objectionable. This objection was overcome by the use of magnesium oxide which is only slightly less effective than the sulphate and produces no by-product. Our work indicates that, while the ionic magnesium, in the form of the bicarbonate of

sulphate, is somewhat more effective than the magnesium oxide, the difference is not as great as that indicated by the author.

In Fig. 13 the curves *C* and *D* represent the difference on the basis of tests conducted on a natural water having a silica content of 44 ppm. Curve *D* represents results obtained by the use of magnesium sulphate, and *C* by the use of a commercial magnesium oxide. Curve *F* represents results with the use of magnesium sulphate on a water having an initial silica content of 30 ppm. Curves *A* and *B* are plotted from points taken from the author's paper, Fig. 9.

While the author's results showing the effectiveness of ionic magnesium (Fig. 1, curve for 90 C) agree closely with our own findings with magnesium sulphate, shown by curves *F* and *D*, Fig. 13, his data on the use of magnesium oxide indicate the necessity of using a quantity far in excess of that required by our research. This difference is indicated by a comparison of curves *F* and *B*, representing tests on a water having an initial silica content of 30 ppm. The difference in results can only be explained by the difference in the physical or chemical characteristics of the calcined magnesite which the author used and the magnesium oxide we employed. In view of the fact that the magnesium oxide we used is commercially available, it is obvious that the process can be much improved by proper selection of the reagents employed.

From the author's work with the absorption of silica in the presence of the phosphate ion, as illustrated in Fig. 9 of the paper, he comes to the conclusion, "the presence of phosphate ion interferes with silica absorption by magnesium oxide." His conclusions are not borne out by our own researches in the laboratory or observations made with hot-process water-softener installations. As an illustration, Fig. 13 of this discussion reproduces the author's findings in which curve *B* illustrates the reduction of silica obtained by the use of calcined magnesite, while curve *A*

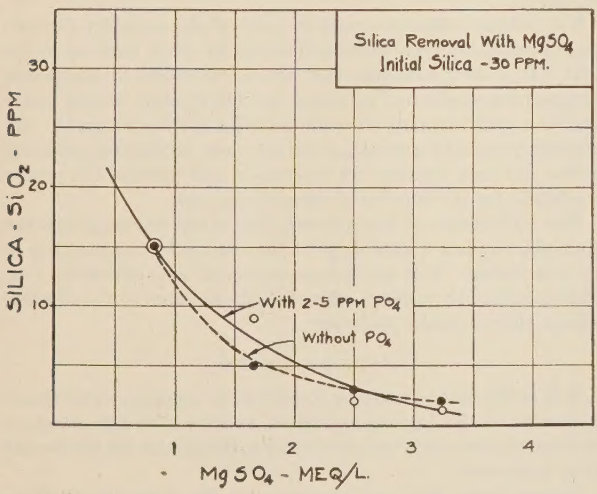


Fig. 14 Silica Removal With $MgSO_4$ With and Without PO_4 Ion (Initial silica, 30 ppm.)

illustrates the same procedure, excepting that precipitation is conducted in the presence of approximately 3 ppm of phosphate ion. On a water containing 30 ppm of silica, which is the concentration the author uses for his explanation, we obtain by the use of magnesium sulphate with the presence of 2 to 5 ppm PO_4 a curve illustrated by *E*, and *F* illustrates the same procedure without the phosphate ion being present. It is quite evident from these curves that, while there may be a slight difference in the absorbing properties of magnesium-hydrate precipitate, it is quite evident that the author's differences cannot be reconciled with our own work.

Curves *E* and *F* resulted from gravimetric-silica tests, whereas, these same observations were made using colorimetric-silica procedures, and are illustrated in Fig. 14 of this discussion. It will be observed that their points are so close it would be impossible to draw the conclusion that the phosphate ion interferes with the absorbing properties of magnesium hydrate.

The reason for this may be in the type of reagents employed for it is quite clear from Fig. 13, with the comparison of curves *B* and *F*, that the author's efficiency of removal is very much less than our own findings. This cannot be ascribed to the differences existing between the use of magnesium oxide and ionic magnesium sulphate, as these differences are illustrated clearly by curves *C* and *D*.

Observation in the field verifies our research that, within the limits of concentrations of magnesium oxide and PO_4 ion used in practice, the presence of the PO_4 ion does not interfere perceptibly with the silica-removal process.

The author points out the effectiveness of sludge recirculation which is a substantiation of our own researches and practical experience in the field, in which the magnesium consumption can be reduced to approximately 50 per cent in normal cases by employing such means.

While he does not present in his paper the effect of magnesium oxide on the absorption of the phosphate ion, this subject has been brought up previously and used as a reason for conducting a silica-removal and phosphate-softening process in two stages. Analysis of the quantity of phosphate absorbed by the magnesium precipitate does not substantiate a sound basis for this recommendation. Our observations have indicated that when 5 ppm of phosphate ion are introduced as an excess, there is no absorption with the use of 50 ppm magnesium sulphate, absorption of only 1 ppm with the use of 100 ppm of magnesium sulphate, and 3 ppm with the use of 150 ppm. In view of the fact that recirculation of sludge is customarily employed in all processes, the quantity of magnesium salts necessary for the normal silica reduction will not generally exceed 100 ppm of magnesium sulphate, or its equivalent in magnesium oxide. Therefore, it does not appear to be economically justified to complicate hot-process softening apparatus merely for the saving of 1 ppm of phosphate.

Our work in the laboratory and in the field does not indicate that the silica-removal process need be separated from that of phosphate softening wherever the phosphate-softening process is applicable. The separation of these processes, as recommended by the author, would add unnecessarily to the equipment cost and attendance.

S. E. Tray.[11] Current interest in methods of reducing the silica content of water used for boiler feed is not only a tribute to the advancement of the science of water treatment, but is indicative of the important role which research plays in the development of steam and power generation. The author is to be commended on the manner in which he has presented data which will prove an important contribution in the field of silica reduction. It is interesting to note that, while the effect of magnesium on the reduction of soluble silica has been quite well known for a number of years, there has been little effort to investigate all of the possibilities of the process until comparatively recently. It is also interesting to observe the number of investigators simultaneously engaged in the study of this process, which only serves to emphasize the importance of the subject.

The author makes an interesting observation on the subject of "silica tolerance," which is the maximum allowable amount of

[11] Allis-Chalmers Manufacturing Company, Milwaukee, Wis.

silica that may be tolerated in the concentrated boiler saline. From the standpoint of silica-scale deposits in boilers, it would appear that a minimum silica content of boiler water is as desirable as minimum alkalinity and total solids. The usual boiler-water sample taken from a boiler drum is assumed to represent average boiler-water conditions. Unfortunately, there have been no means developed thus far to obtain water samples from tubes in which the maximum heat transfer occurs. As a result, we are somewhat in the dark as to the actual boiler-water conditions at these points of greatest evaporation. It is entirely possible for the silica to remain in solution throughout most of the boiler, and yet to precipitate in the solid phase at these localized points. Until we have a better understanding of the conditions under which silica precipitates, it would seem wise to reduce the silica concentration to the lowest possible value.

From the standpoint of turbine operation, a study of a large number of plants indicates that if insoluble silica deposits on turbine blades are to be avoided, the soluble silica in the boiler water must not exceed 2 per cent of the total dissolved solids. From this standpoint as well, it is very desirable to reduce the soluble silica as low as possible.

The results reported by the author are apparently based on laboratory tests, rather than full-scale plant operation. A note of caution should be sounded against attempting to apply laboratory results directly to plant practice. The author has indicated that the reduction of silica is largely a physical process and, thus, the concentrations and character of the sludge are of prime importance. It is extremely difficult to reproduce in the laboratory the conditions which normally exist in large-scale operations. It would be interesting also to have more detailed information on the chemical characteristics of the effluent following treatment for silica reduction. The hardness and alkalinity of the effluent, as well as the control required to obtain maximum silica removal, are important considerations from the standpoint of practical operation.

This paper points out the effect of ionic magnesium on silica removal, and indicates this source of magnesium to be more efficient and to require less than magnesium derived from sludge. This is not entirely true, since the ionic magnesium must precipitate from solution, usually as the hydroxide, in order to remove silica. In the process of precipitation, the magnesium passes from the soluble to the insoluble state through a colloidal phase, and the particle size of the precipitate during this process ranges from zero to a definite, measurable size. The paper indicates that there is a particle size of sludge which is more effective in silica removal than any other, and obviously, in the case of ionic magnesium, this state must be reached some time during the process. However, undissolved magnesium compounds may also exist in this favorable state, provided a suitable form of magnesium is selected and the proper conditions are maintained. With these requirements satisfied, the undissolved magnesium compounds may be equally as effective in removing silica, without the necessity of recycling the sludge.

There is one inherent disadvantage in the magnesium dissolver which tends to reduce its effectiveness as a source of ionic magnesium. As the sludge is recycled and brought in contact with low pH water, $Mg(OH)_2$ will dissolve first. However this sludge also contains absorbed silica, the same as the waste sludge. This silica will also dissolve in low pH water and, hence, the SiO_2 content of the incoming water will be increased. As a consequence, the magnesium requirements will continue to increase in proportion to the silica added from the recycled sludge.

The author indicates that the removal of silica by magnesium is generally an adsorption process, but also states that it is possible this process involves other phenomena besides straight adsorption. Our experience confirms this thought, which is also borne out by the fact that the efficiency of the process definitely increases with temperature. In a true adsorption process, increased temperature reduces the rate of adsorption. The author finds that, as the ratio of $Mg:SiO_2$ rises in the sludge, the residual silica in the effluent decreases. This would be true in the case of straight adsorption, but experience in the field shows that the $Mg:SiO_2$ ratio in the sludge does not necessarily determine the degree of silica removal. The state of the sludge material seems to be quite important, and this likewise is contrary to what we would expect of straight adsorption. It is also inferred that if magnesium sludge were left indefinitely in the sludge zone, silica would be adsorbed indefinitely. After the sludge has reached equilibrium there will be no further removal of silica from solution.

Adsorption may be differentiated from precipitation by the fact that in an adsorption process no definite compound can be identified. It has been possible, however, by means of X-ray-diffraction analysis to identify certain combinations of magnesium and silica in the precipitated sludge. It is more likely, therefore, that removal of silica by magnesium is a combination of both adsorption and precipitation. The factors favorable to both phenomena must be maintained in order to obtain the maximum effectiveness of the treatment.

The effect of phosphate on silica removal with magnesium is well known, and it is interesting to learn that the tests reported by the author confirm this point. The fact that the phosphate ion interferes with the action of magnesium on silica is due in part to the respective solubilities of magnesium phosphate and magnesium silicate. It is also due to the blanketing effect of the phosphate ion, which is adsorbed by the magnesium along with the silica. This reduces the amount of silica which can be adsorbed by a given amount of magnesium. There is no doubt that the removal of silica, even from low-hardness waters, must be accomplished prior to the addition of the phosphate ion for softening.

The author makes an excellent point of the necessity for intimate contact between water and sludge for silica removal in the cold. It is very fortunate that the development of equipment designed for accelerated softening and clarification in cold water has been contemporary with the progress in silica removal. By providing this intimate sludge contact, these accelerated softeners reduce the time required for adsorption and provide the proper conditions for the removal of silica in the cold.

The application of this process, like other water-purification methods, requires a knowledge of the chemical characteristics of the raw water. It is not a panacea for all silica troubles; but, when intelligently applied, offers definite assistance in combating soluble silica in boiler feedwater.

Author's Closure

It is particularly gratifying to have such copious and pertinent discussion on this important subject, because it directs attention to those phases which require further investigation for the benefit of all concerned.

Messrs. Betz, Noll, and Maguire, and Mr. Joos call attention to the fact that the undissolved magnesium oxides on which data were presented in the paper do not appear to be as absorptive as other MgO products on the market. In this connection, it should be borne in mind that the primary focus of this entire paper is on various methods of silica removal which can be combined advantageously under various conditions to produce maximum silica removal *at minimum cost.* We were not attempting to present data on high-cost activated MgO products as we believe such data have been presented elsewhere by these discussers and some time previously by other investigators such as Wesly (see reference 42 in the Bibliography). This low-cost point of view is verified by the fact that Table 1 of the paper

states a price of 3 cents per pound for MgO, whereas the activated MgO products to which the discussers refer are marketed at more than double this price.

Another point on which there seems to be considerable confusion is the "magnesium dissolver" and its role in combining various methods of silica removal in an efficient low-cost process. Messrs. Betz, Noll, and Maguire, and Mr. Crossan seem to feel that this dissolver is not usually needed and that it adds undue complication to the operation of the plant. There is one important point to bear in mind in this connection, and that is that, so long as no CO_2 is added to the influent from any external source such as flue gas, the use of the magnesium dissolver does not involve either any increased operating cost or any additional operating control. This is fully discussed in the paper under the heading "Combined Use of Ionic Magnesium and Undissolved Magnesium Compounds," on page 55.

Mr. Tray has raised a question about the increase in silica content upon passing recirculated sludge through the magnesium dissolver. Actual tests show that the silica content of the treated water leaving the dissolver is not higher than that of the influent, while the use of the dissolver, as proved by experience, does reduce the final silica content without additional operating cost.

There seems to be considerable difference of opinion on the effect of PO_4 ion. These opinions are apparently based on different test results rather than on different interpretations of similar data. Messrs. Betz, Noll, and Maguire, and Mr. Joos, feel that the PO_4 ion does not have any significant effect, while Mr. Tray agrees with us that it does. It is pertinent to mention here that previous investigators (Wesly, references 13 and 42 of the paper) also find that PO_4 ion interferes with silica absorption by MgO. These conflicting data indicate that either there are some incorrect measurements involved or that different waters act differently in this respect. It is to be hoped that further parallel tests on various waters will clear up this question in due time.

Mr. Tray questions the "silica tolerance" and suggests the advisability of maintaining the SiO_2 concentration as low as possible in the boiler salines. There is much to be said for his point of view. Furthermore, it makes it all the more desirable to combine all the advantages of ionic Mg precipitation, high Mg sludge concentration, recycling sludge through the magnesium dissolver, and utilizing the cheapest possible source of magnesium to obtain the best results at lowest operating cost. This is particularly true in those cases where the quantity of water to be treated is large, so that the operating cost per unit of water treated is important.

Some questions are also raised about the fact that this paper is based on laboratory tests. Although this point is well taken, it should be noted that special laboratory techniques were developed (as described in the paper) to simulate long-term, large-scale tests, and these techniques were verified by operating larger pilot plant equipment over a long period. As a matter of fact, Mr. Crossan points out that one modification of our dolomitic lime process has been in successful operation at his plant for over one and a half years. It may be added that considerable further experience has become available since this paper was presented about a year ago, and these large-scale experiences further verify the conclusions drawn from the tests presented in the paper. It is intended to present these operating data in another paper in the near future.